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HEAT TRANSFER IN AMMONIA CONDENSERS

PART III

BY

ALONZO P. KRATZ
HORACE J. MACINTIRE
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ENGINEERING EXPERIMENT STATION

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HEAT TRANSFER IN AMMONIA CONDENSERS

PART III

BY

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HEAT TRANSFER IN AMMONIA CONDENSERS

PART III

I. Introduction

1. *Preliminary Statement.*—This bulletin constitutes a report of the progress made in the investigation of the heat transfer in various types of ammonia condensers since the publication of Engineering Experiment Station Bulletins Nos. 171 and 186. The former dealt with the performance of three types of ammonia condensers, namely, the atmospheric-bleeder, the double-pipe, and the vertical shell-and-tube; the latter with results obtained when certain changes were made in the design and operation of the vertical shell-and-tube condenser and with results from a study of the heat transfer in a double-tube superheat remover. The present bulletin presents results from a study of the performance of a horizontal shell-and-tube condenser, sometimes designated as the multitube-multipass type, over a wide range of operating conditions, and with certain variations in the arrangement of surfaces.

2. *Objects of Investigation.*—The objects of this investigation may be stated briefly as follows:

(1) To determine the coefficient of heat transfer and performance characteristics of the condenser with the water passing in parallel through the two shells and with various rates of flow and initial temperatures for the water.

(2) To determine the effect of the pressure of the ammonia in the condenser on the coefficient of heat transfer and the performance characteristics of the condenser.

(3) To determine the performance characteristics when the condenser was operated with one shell alone, and also with the water passing in series through the two shells instead of in parallel.

(4) To determine the rate of scale formation, or fouling of the tubes, and the effect of such fouling action on the coefficient of heat transfer for the condenser.

3. *Acknowledgments.*—This investigation has been part of the work of the Engineering Experiment Station of the University of Illinois, of which Dean M. S. Ketchum is the director, and of the Department of Mechanical Engineering of which Prof. A. C. Willard is the head.

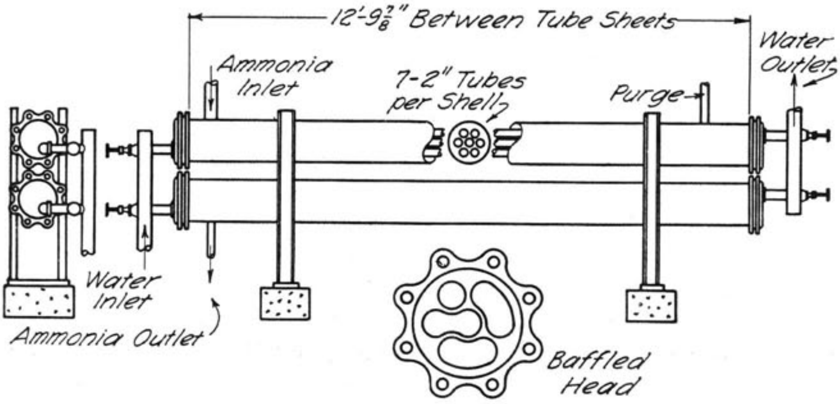

FIG. 1. DIAGRAM OF HORIZONTAL SHELL-AND-TUBE CONDENSER

II. DESCRIPTION OF APPARATUS

4. *Condenser.*—The condenser used for these tests was of the multitube-multipass type, as shown in Fig. 1, and consisted of two shells, each shell containing seven tubes 12.82 ft. long. The tubes were made of charcoal iron, and each tube had an outside diameter of 2.00 in. and an inside diameter of 1.81 in. The condenser had a total of 93.8 sq. ft. of condensing surface. The tubes were arranged in the shell so that the water passed through the seven tubes in series, the ends of the tubes being connected by return bends integral with the condenser heads. The two shells were connected in parallel, and equal amounts of water passed through the tubes in each shell. Each shell was lagged with 2 inches of magnesia block insulation. The superheat was removed from the ammonia gas in a separate superheat remover and the ammonia entered the condenser in a practically dry saturated condition.

5. *Testing Apparatus.*—The amount of ammonia condensed was weighed in two drums placed on scales. The connecting piping had sufficient flexibility so that the scales were sensitive to a change of 0.1 lb. in the weight on the scale platforms. The cooling water was weighed in tanks on platform scales. This apparatus, together with the plant and the thermocouple system used for obtaining temperatures, has been described in detail in Bulletin No. 171.

All temperatures were obtained by means of copper-constantan thermocouples of No. 22 B.&S. gage wire. The temperature of the ammonia vapor was observed at ten points in each shell, and of the water at the points of entry at each tube in the shell and at the point of exit for the last tube. The thermocouples in the ammonia vapor were protected by encasing them in nickel p ated copper tubing, and those in the water were placed inside of glass tubes filled with oil. In all cases a depth of immersion sufficient to eliminate the effect of conduction along the thermocouple leads was employed.

III. Method of Procedure

6. *General Method for Conducting Tests.*—The general method for conducting tests and controlling conditions has been fully described in Bulletin No. 171.

7. *Test Conditions for the Various Series.*—The different series of tests were each designated by a letter. For the F series the ammonia liquefaction pressure was maintained at 145 lb. per sq. in. gage and tests were run for six different rates of condensation varying from approximately 2.8 to 7.0 lb. of ammonia condensed per minute, and corresponding to condenser tonnages of from 5.68 to 17.24. For each rate of condensation six different water rates were used for the cooling water, varying from 85 to 350 lb. per min. In all cases both shells of the condenser were used and the water passed through the two shells in parallel.

All tests were run with commercially clean tubes except test XF for which the tubes were purposely allowed to become fouled. Considerable difficulty was experienced from the tubes becoming fouled with a very soft scale-like deposit and it was necessary to clean them after approximately 6 hours of running. The temperature difference between the ammonia and the water was closely observed, however, and testing was discontinued at the first evidence of fouling.

The G and H series were run under the same conditions as the F series except that for the G series the condenser pressure was maintained at 172 lb. per sq. in. gage, and for the H series at 122 lb. per sq. in. gage. For the G series three and for the H series four different rates of condensation were used.

For the I series only one shell of the condenser was effective, the second shell being completely removed from service. The tests were run under the same conditions as for the F series, with the pressure maintained at 145 lb. per sq. in. gage. Three different rates of condensation were used, and the water rate was varied from 120 to 375 lb. per min.

In the case of the J series the condenser was re-piped so that the water passed first through one shell and then through the second shell in series. All tests were run with a condenser pressure of 145 lb. per sq. in. gage, and three rates of condensation were used. The water rates were varied as in the cases for the other series.

The GX series was run for the purpose of determining the rate of scale formation. For these tests the condenser was operated continuously over a long period of time at a constant rate of condensation and a constant water rate, and observations were made at approximately one-hour intervals of the mean temperature difference between the ammonia vapor and the water. The increase in this temperature difference was an indication of the rate at which fouling occurred. A rate of condensation of approximately 4.7 lb. of ammonia per min. was used for both tests in the series. A water rate of 209 lb. per min. was used for one test and of 331 lb. per min. for the other.

8. *Method of Calculation.*—The formulas for computing the tests were developed in Bulletin No. 171.

The condenser tonnage was computed from the formula

$$T = \frac{N(i'' - i')}{200}$$

and the coefficient of heat transfer from the formula

$$K = \frac{60N(i'' - i')}{A\ \theta_m}$$

in which T = condenser tonnage

N = ammonia condensed, lb. per min.

i'' = heat content of dry saturated ammonia vapor at the temperature of liquefaction in the condenser, B.t.u. per lb.

i' = heat content of the liquid at the temperature of the liquid leaving the condenser, B.t.u. per lb.

K = average coefficient of heat transfer, or the B.t.u. transmitted per sq. ft. per hr. per deg. difference in temperature.

A = effective condensing surface, sq. ft.

θ_m = mean temperature difference between the ammonia and the cooling water, deg. F.

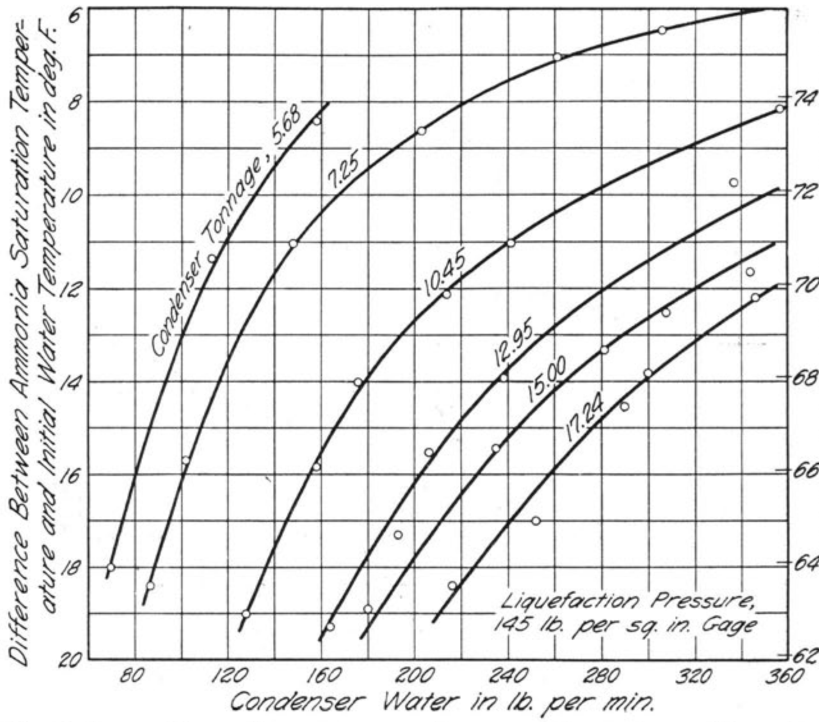

FIG. 2. INITIAL WATER TEMPERATURE FOR DIFFERENT TONNAGES AND WATER RATES IN F SERIES OF HORIZONTAL SHELL-AND-TUBE CONDENSER TESTS

In the case of these tests the value of θ_m was computed by averaging the temperatures determined from the readings of the thermocouples in the ammonia, and also those from the readings of the thermocouples in the water, and taking the difference between the two averages, since this method gave consistent results.

IV. RESULTS OF TESTS

9. *General.*—The principal results from all tests are given in Table 1. Each test has been assigned a letter representing the series to which it belongs, and a number representing the serial number of the test itself with respect to the series. The conditions which were established for the different series are discussed in Section 7.

10. *Total Effective Cooling Water and Total Condenser Tonnage.*—The relations between the initial temperature of the cooling water, the total weight of the cooling water circulated and the total condenser tonnage are shown in the curves of Figs. 2 to 4. These curves

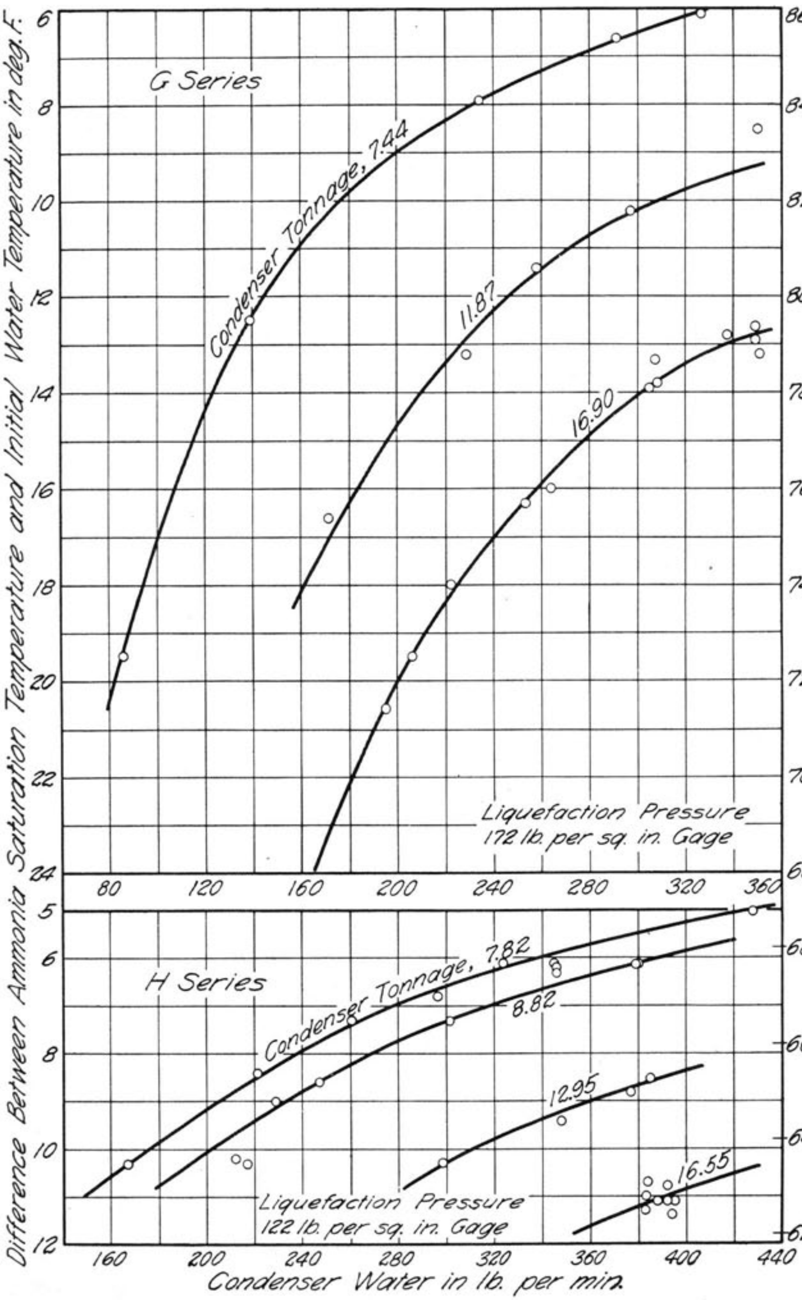

FIG. 3. INITIAL WATER TEMPERATURE FOR DIFFERENT TONNAGES AND WATER RATES IN G AND H SERIES OF HORIZONTAL SHELL-AND-TUBE CONDENSER TESTS

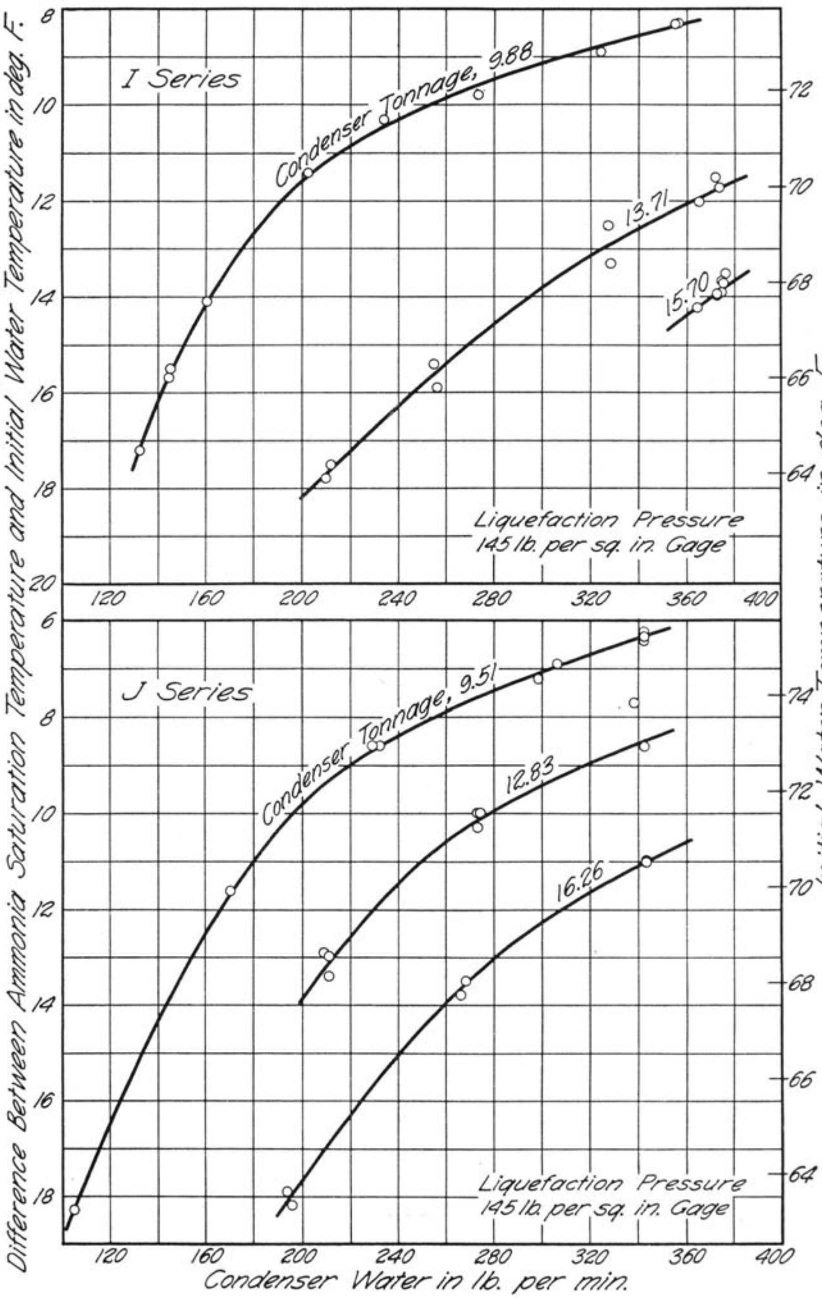

FIG. 4. INITIAL WATER TEMPERATURE FOR DIFFERENT TONNAGES AND WATER RATES IN I AND J SERIES OF HORIZONTAL SHELL-AND-TUBE CONDENSER TESTS

TABLE 1

PRINCIPAL RESULTS OF TESTS ON HORIZONTAL SHELL-AND-TUBE CONDENSER

Test Number	Ammonia lb. per min.	Condenser Tonnage	Average Ammonia Saturation Temp. deg. F.	Diff. between Ammonia Saturation Temp. and Initial Water Temp. deg. F.	Initial Water Temp. deg. F.	Final Water Temp. deg. F.	Average Water Film Temp. deg. F.	Mean Temp. Drop through Ammonia Film deg. F.	Mean Temp. Drop through Tube Wall deg. F.	Mean Temp. Diff. between Ammonia and Water, deg. F.	Coefficient of Heat Transfer B.t.u. per hr. per sq. ft. per deg. F.	Condenser Water lb. per min.	Condenser Water gal. per min. per sq. ft.	Condenser Water ft. per sec.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
1F	2.96	7.40	81.3	18.4	62.9	79.8	77.5	0.58	0.28	7.11	133	87	0.114	0.65
2F	2.90	7.25	80.9	15.7	65.2	79.4	77.3	0.57	0.22	6.37	146	102	0.131	0.76
3F	2.80	7.00	81.2	11.0	70.2	80.6	78.3	0.55	0.21	4.98	180	148	0.189	1.11
4F	2.86	7.15	82.0	8.6	73.4	80.5	79.5	0.56	0.22	4.15	221	203	0.260	1.52
5F	2.94	7.35	82.1	7.0	75.1	80.6	79.9	0.57	0.22	3.54	266	261	0.334	1.95
6F	2.95	7.37	82.5	6.4	76.1	81.0	80.1	0.58	0.22	3.34	283	296	0.379	2.22
7F	6.93	17.32	82.3	13.8	68.5	80.2	77.8	1.35	0.53	7.12	311	300	0.384	2.24
8F	6.86	17.16	81.9	12.1	69.8	79.8	77.2	1.34	0.52	6.38	344	346	0.443	2.59
9F	7.00	17.50	82.1	14.5	67.6	80.0	77.5	1.37	0.58	7.34	305	290	0.371	2.17
10F	7.04	17.60	82.4	17.0	65.4	79.9	77.2	1.87	0.53	8.40	268	252	0.323	1.88
11F	6.64	16.60	82.0	18.4	63.6	79.5	76.6	1.30	0.50	8.91	238	216	0.276	1.62
12F	5.82	14.55	82.1	18.9	63.2	79.7	76.9	1.13	0.44	8.76	213	180	0.230	1.35
13F	6.04	15.10	82.3	15.4	66.9	80.0	77.7	1.18	0.46	7.50	258	235	0.301	1.76
14F	6.06	15.15	82.4	13.3	69.1	80.1	76.2	1.18	0.46	6.84	284	281	0.360	2.10
15F	6.12	15.30	82.3	12.5	69.8	80.0	78.2	1.19	0.47	6.54	299	308	0.394	2.30
16F	5.97	14.93	81.9	11.6	70.3	79.6	78.0	1.17	0.45	6.14	311	344	0.440	2.58
17F	4.26	10.63	82.2	19.0	63.2	81.2	78.0	0.83	0.32	7.28	187	128	0.164	0.96
18F	4.07	10.18	81.7	15.8	65.9	81.0	77.8	0.79	0.31	6.65	196	158	0.202	1.18
19F	4.01	10.03	81.7	14.0	67.7	81.0	78.1	0.78	0.30	6.02	213	176	0.225	1.32
20F	4.30	10.75	81.9	12.1	69.8	81.2	78.6	0.84	0.33	5.37	256	214	0.274	1.60
21F	4.15	10.38	81.7	11.0	70.7	81.0	78.5	0.81	0.32	5.07	262	241	0.308	1.80
22F	4.30	10.75	82.4	8.1	74.3	81.7	79.8	0.84	0.33	4.03	341	357	0.457	2.67
23F	5.30	13.27	81.7	19.3	62.4	80.3	76.6	1.03	0.40	8.70	195	163	0.208	1.22
24F	5.22	13.05	81.3	17.3	64.0	79.9	76.6	1.02	0.40	8.00	209	193	0.247	1.44
25F	5.04	12.60	82.0	15.5	66.5	80.6	77.7	0.98	0.38	7.14	226	206	0.264	1.54
26F	5.22	13.05	81.7	13.9	67.8	80.5	77.7	1.02	0.40	6.53	256	239	0.306	1.79
27F	5.12	12.80	82.0	9.7	72.3	80.6	78.8	1.00	0.39	5.07	323	337	0.431	2.52
28F	2.26	5.65	82.2	18.0	64.2	80.7	78.4	0.44	0.17	7.04	103	70	0.090	0.52
29F	2.20	5.50	82.0	8.4	73.6	80.9	79.8	0.43	0.17	3.80	185	159	0.204	1.19

TABLE 1 (Continued)

PRINCIPAL RESULTS OF TESTS ON HORIZONTAL SHELL-AND-TUBE CONDENSER

Test Number	Ammonia lb. per min.	Condenser Tonnage	Average Ammonia Saturation Temp. deg. F.	Diff. between Ammonia Saturation Temp. and Initial Water Temp. deg. F.	Initial Water Temp. deg. F.	Final Water Temp. deg. F.	Average Water Film Temp. deg. F.	Mean Temp. Drop through Ammonia Film deg. F.	Mean Temp. Drop through Tube Wall deg. F.	Mean Temp. Diff. between Ammonia and Water, deg. F.	Coefficient of Heat Transfer B.t.u. per hr. per sq. ft. per deg. F.	Condenser Water lb. per min.	Condenser Water gal. per min. per sq. ft.	Condenser Water ft. per sec.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
30F	2.36	5.90	82.2	11.4	70.8	81.0	79.5	0.46	0.18	4.84	156	113	0.145	0.85
31F	3.58	8.95	80.6	15.3	65.3	79.2	76.9	0.74	0.27	6.45	178	130	0.166	0.97
32F	3.43	8.57	81.3	5.8	75.5	80.0	79.3	0.67	0.26	3.07	357	379	0.485	2.67
XF	3.09	7.73	81.5	8.3	73.2	78.4	79.4	0.60	0.24	5.28	187	294	0.376	2.20
YF	3.07	7.68	82.3	6.2	76.1	81.0	80.2	0.60	0.24	3.26	301	300	0.384	2.25
1G	3.09	7.52	92.0	6.1	85.9	91.1	90.1	0.59	0.20	3.03	318	327	0.418	2.44
2G	3.12	7.58	91.6	6.6	85.0	90.7	89.6	0.59	0.21	3.26	298	291	0.372	2.18
3G	3.19	7.75	92.1	7.9	84.2	91.2	89.9	0.61	0.21	3.67	270	234	0.299	1.75
4G	3.00	7.29	91.7	9.9	81.8	90.8	89.2	0.57	0.20	4.23	221	179	0.229	1.34
5G	3.00	7.29	91.9	12.5	79.4	91.0	89.0	0.57	0.20	4.93	189	139	0.178	1.04
6G	2.95	7.17	91.9	19.5	72.4	91.0	88.1	0.56	0.20	6.91	133	86	0.110	0.64
7G	6.97	16.92	92.5	27.2	65.3	88.9	85.5	1.32	0.46	12.20	178	144	0.184	1.08
8G	7.15	17.35	92.2	19.5	72.7	88.6	86.5	1.36	0.47	9.62	231	206	0.264	1.54
9G	6.82	16.57	92.3	16.3	76.0	89.0	87.3	1.30	0.45	8.27	256	253	0.324	1.89
10G	7.05	17.10	91.5	12.8	78.7	88.8	87.3	1.34	0.47	6.67	329	337	0.431	2.52
11G	7.04	17.09	91.5	13.9	77.6	88.6	87.0	1.34	0.47	7.12	307	305	0.391	2.28
12G	4.60	11.18	92.1	8.5	83.6	90.3	89.3	0.87	0.30	4.50	318	350	0.448	2.62
13G	5.07	12.30	91.4	10.2	81.2	89.4	88.2	0.97	0.34	5.17	304	297	0.380	2.22
14G	4.80	11.66	92.8	11.4	81.4	90.6	89.3	0.91	0.32	5.75	260	258	0.330	1.93
15G	4.98	12.10	92.2	13.2	79.0	90.0	88.4	0.95	0.33	6.24	248	229	0.293	1.71
16G	4.98	12.10	92.3	16.6	75.7	90.1	88.1	0.95	0.33	7.17	216	171	0.219	1.28
17G	7.15	17.35	91.8	13.2	78.6	88.4	87.3	1.36	0.47	7.19	310	351	0.450	2.62
18G	6.91	16.78	92.0	12.9	79.1	89.2	87.7	1.32	0.46	6.89	312	349	0.447	2.61
19G	6.85	16.62	91.8	12.6	79.2	89.1	87.6	1.30	0.45	6.72	317	349	0.447	2.61
20G	6.80	16.50	92.3	13.3	79.0	89.6	87.9	1.29	0.45	7.04	300	307	0.393	2.29
21G	6.88	16.70	91.9	13.8	78.1	89.2	87.4	1.31	0.45	7.18	298	308	0.394	2.30
22G	6.98	16.95	92.6	16.0	76.6	89.8	87.8	1.33	0.46	7.86	276	264	0.338	1.97

TABLE 1 (Continued)

PRINCIPAL RESULTS OF TESTS ON HORIZONTAL SHELL-AND-TUBE CONDENSER

Test Number	Ammonia lb. per min.	Condenser Tonnage	Average Ammonia Saturation Temp. deg. F.	Diff. between Ammonia Saturation Temp. and Initial Water Temp. deg. F.	Initial Water Temp. deg. F.	Final Water Temp. deg. F.	Average Water Film Temp. deg. F.	Mean Temp. Drop through Ammonia Film deg. F.	Mean Temp. Drop through Tube Wall deg. F.	Mean Temp. Diff. between Ammonia and Water, deg. F.	Coefficient of Heat Transfer B.t.u. per hr. per sq. ft. per deg. F.	Condenser Water lb. per min.	Condenser Water gal. per min. per sq. ft.	Condenser Water ft. per sec.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
23G	6.81	16.53	92.3	18.0	74.3	89.5	87.1	1.30	0.45	8.61	246	222	0.284	1.66
24G	7.06	17.12	91.4	20.6	70.8	88.6	85.8	1.34	0.47	9.47	232	195	0.250	1.46
1H	2.97	7.50	74.1	10.3	63.8	72.8	71.3	0.58	0.23	4.69	205	167	0.214	1.25
2H	3.10	7.84	74.1	8.4	65.7	72.9	71.7	0.61	0.24	3.97	252	221	0.283	1.65
3H	3.14	7.94	74.3	7.3	67.0	73.2	72.1	0.62	0.24	3.54	286	260	0.333	1.94
4H	3.12	7.88	74.2	6.8	67.4	73.1	72.1	0.61	0.24	3.37	299	296	0.379	2.22
5H	3.06	7.73	74.3	6.1	68.2	73.2	72.3	0.60	0.23	3.19	310	324	0.415	2.42
6H	3.19	8.06	73.9	5.0	68.9	72.9	72.1	0.63	0.24	2.76	374	428	0.548	3.20
7H	6.58	16.60	73.6	11.1	62.5	71.2	69.6	1.30	0.50	6.10	349	392	0.501	2.93
8H	6.66	16.80	73.7	11.1	62.6	71.3	69.7	1.31	0.51	6.11	352	388	0.496	2.90
9H	6.55	16.53	73.8	11.1	62.7	71.4	69.9	1.29	0.50	6.04	353	395	0.505	2.96
10H	6.44	16.26	73.2	10.8	62.4	70.9	69.4	1.27	0.49	5.90	352	392	0.501	2.93
11H	6.73	17.00	73.3	11.4	61.9	70.9	69.3	1.33	0.51	6.23	348	394	0.504	2.94
12H	6.38	16.10	73.0	10.7	62.3	70.7	69.2	1.25	0.49	5.81	354	384	0.491	2.87
13H	6.40	16.15	73.0	11.0	62.0	70.6	69.1	1.26	0.49	6.04	342	383	0.490	2.86
14H	6.68	16.87	73.2	11.3	61.9	70.7	69.1	1.32	0.51	6.30	342	383	0.490	2.86
15H	3.28	8.28	73.3	6.1	67.2	72.3	71.3	0.65	0.25	3.04	349	345	0.441	2.58
16H	3.42	8.63	73.6	6.2	67.4	72.6	71.5	0.67	0.26	3.17	348	346	0.442	2.59
17H	3.48	8.79	73.4	6.3	67.1	72.2	71.3	0.69	0.27	3.31	340	346	0.442	2.59
18H	3.58	9.05	73.4	9.0	64.4	72.1	70.8	0.70	0.27	4.30	269	229	0.283	1.71
19H	3.61	9.12	73.8	6.1	67.7	72.3	71.6	0.71	0.28	3.43	340	380	0.486	2.84
20H	3.48	8.79	74.1	6.1	68.0	72.6	71.9	0.68	0.27	3.40	331	379	0.485	2.83
21H	3.50	8.84	74.0	7.3	66.7	72.4	71.6	0.69	0.27	3.91	289	291	0.372	2.18
22H	3.44	8.69	74.1	8.6	65.5	72.3	71.4	0.68	0.26	4.45	250	247	0.316	1.85
23H	3.62	9.14	74.6	10.3	64.3	72.6	71.5	0.71	0.28	5.23	224	217	0.278	1.62
24H	3.53	8.92	73.9	10.2	63.7	72.0	70.8	0.69	0.27	5.16	221	212	0.272	1.58
25H	5.22	13.19	74.2	8.8	65.4	72.0	71.1	1.03	0.40	4.81	351	377	0.482	2.82
26H	5.15	13.00	74.1	8.5	65.6	72.2	71.1	1.01	0.39	4.63	359	385	0.493	2.88

TABLE 1 (Continued)

PRINCIPAL RESULTS OF TESTS ON HORIZONTAL SHELL-AND-TUBE CONDENSER

Test Number	Ammonia lb. per min.	Condenser Tonnage	Average Ammonia Saturation Temp. deg. F.	Diff. between Ammonia Saturation Temp. and Initial Water Temp. deg. F.	Initial Water Temp. deg. F.	Final Water Temp. deg. F.	Average Water Film Temp. deg. F.	Mean Temp. Drop through Ammonia Film deg. F.	Mean Temp. Drop through Tube Wall deg. F.	Mean Temp. Diff. between Ammonia and Water, deg. F.	Coefficient of Heat Transfer B.t.u. per hr. per sq. ft. per deg. F.	Condenser Water lb. per min.	Condenser Water gal. per min. per sq. ft.	Condenser Water ft. per sec.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
27H	5.15	13.00	74.2	9.4	64.8	72.0	70.9	1.01	0.39	5.17	322	348	0.445	2.60
28H	4.99	12.60	74.5	10.3	64.2	72.3	71.1	0.98	0.38	5.44	296	298	0.381	2.23
1I	5.46	13.64	81.4	11.5	69.9	77.2	77.4	0.53	0.21	7.33	472	372	0.953	5.56
2I	5.31	13.28	81.5	11.7	69.8	77.2	77.4	0.52	0.20	7.48	454	373	0.955	5.58
3I	5.45	13.63	82.0	12.0	70.0	77.7	77.8	0.53	0.21	7.64	457	365	0.935	5.46
4I	5.65	14.12	81.9	13.3	68.6	77.5	77.4	0.56	0.22	8.20	443	328	0.840	4.91
5I	5.32	13.30	81.3	12.5	68.8	77.2	77.1	0.52	0.20	7.68	443	327	0.837	4.91
6I	5.64	14.11	82.1	15.9	66.2	77.7	77.1	0.55	0.21	9.19	395	256	0.655	3.83
7I	5.52	13.80	81.5	15.4	66.1	77.3	76.7	0.54	0.21	8.88	398	255	0.652	3.82
8I	5.51	13.79	81.6	17.8	63.8	77.3	76.4	0.53	0.21	9.70	362	210	0.538	3.14
9I	5.45	13.63	81.7	17.5	64.2	77.4	76.6	0.35	0.21	9.67	361	212	0.543	3.17
10I	3.54	8.86	81.8	18.2	63.6	78.8	77.2	0.35	0.13	8.71	260	121	0.310	1.81
11I	3.60	9.00	81.5	17.2	64.3	78.4	77.0	0.35	0.14	8.48	273	132	0.338	1.97
12I	3.65	9.14	81.3	15.5	65.8	78.5	77.2	0.36	0.14	7.64	306	145	0.371	2.17
13I	3.55	8.88	81.6	15.7	65.9	79.1	77.5	0.35	0.13	7.76	293	144	0.370	2.16
14I	3.56	8.90	81.3	14.1	67.2	78.5	77.4	0.35	0.13	7.25	314	160	0.409	2.39
15I	3.49	8.72	81.7	11.4	70.3	79.2	78.4	0.35	0.13	6.06	369	202	0.516	3.02
16I	3.44	8.60	81.0	10.3	70.7	78.1	77.8	0.33	0.13	5.88	374	234	0.599	3.50
17I	3.67	9.18	81.7	9.8	71.9	78.8	78.6	0.36	0.14	5.74	409	273	0.699	4.09
18I	3.68	9.20	81.3	8.9	72.4	78.3	78.3	0.36	0.14	5.42	434	324	0.830	4.85
19I	3.61	9.03	81.7	8.3	73.4	78.7	78.8	0.35	0.14	5.25	440	355	0.910	5.31
20I	3.70	9.25	81.9	8.3	73.6	78.9	79.0	0.36	0.14	5.25	451	356	0.911	5.33
21I	6.42	16.05	81.5	14.2	67.3	76.1	76.4	0.63	0.24	9.34	441	364	0.931	5.45
22I	6.12	15.30	81.6	13.5	68.1	76.4	76.7	0.60	0.23	8.88	441	376	0.962	5.62
23I	6.33	15.82	81.9	13.7	68.2	76.8	77.0	0.62	0.24	8.89	456	375	0.960	5.61
24I	6.27	15.68	81.9	13.7	68.2	76.9	77.0	0.61	0.24	8.89	452	375	0.960	5.61
25I	6.29	15.71	82.5	13.9	68.6	77.3	77.5	0.62	0.24	9.11	442	374	0.957	5.60

TABLE 1 (Continued)

PRINCIPAL RESULTS OF TESTS ON HORIZONTAL SHELL-AND-TUBE CONDENSER

Test Number	Ammonia lb. per min.	Condenser Tonnage	Average Ammonia Saturation Temp. deg. F.	Diff. between Ammonia Saturation Temp. and Initial Water Temp. deg. F.	Initial Water Temp. deg. F.	Final Water Temp. deg. F.	Average Water Film Temp. deg. F.	Mean Temp. Drop through Ammonia Film deg. F.	Mean Temp. Drop through Tube Wall deg. F.	Mean Temp. Diff. between Ammonia and Water, deg. F.	Coefficient of Heat Transfer B.t.u. per hr. per sq. ft. per deg. F.	Condenser Water		
												lb. per min.	gal. per min. per sq. ft.	ft. per sec.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
26I	6.22	15.55	82.0	14.0	68.0	76.8	77.0	0.61	0.24	9.11	437	372	0.952	5.56
27I	6.31	15.79	82.0	13.9	68.1	76.8	77.0	0.62	0.24	9.04	447	372	0.952	5.56
1J	3.86	9.65	80.8	6.3	74.5	80.1				2.83	432	342	0.438	5.11
2J	3.83	9.57	80.8	6.4	74.4	79.9				2.94	416	342	0.438	5.11
3J	3.78	9.45	80.6	6.2	74.4	80.1				2.70	448	343	0.439	5.13
4J	3.78	9.45	81.0	6.3	74.7	80.3				2.90	417	343	0.439	5.13
5J	3.79	9.47	81.7	7.2	74.5	81.1				3.06	396	298	0.382	4.46
6J	3.83	9.57	81.7	6.9	74.8	81.1				2.97	412	306	0.392	4.57
7J	3.76	9.40	81.9	8.6	73.3	81.2				3.38	356	232	0.297	3.47
8J	3.70	9.25	81.8	8.6	73.2	81.2				3.34	354	229	0.293	3.43
9J	3.88	9.70	81.8	11.6	70.2	81.3				3.97	312	170	0.218	2.54
10J	3.82	9.55	82.2	18.3	63.9	81.8				5.18	236	105	0.134	1.57
11J	4.68	11.70	81.0	7.7	73.3	80.1				3.54	422	338	0.433	5.06
12J	5.20	13.00	81.5	8.6	72.9	80.5				3.95	421	342	0.438	5.11
13J	5.30	13.25	81.8	10.3	71.5	81.1				4.19	405	273	0.349	4.09
14J	5.00	12.50	81.5	10.0	71.5	80.5				4.23	379	274	0.350	4.10
15J	5.14	12.87	81.7	10.0	71.7	80.8				4.14	397	273	0.349	4.09
16J	5.46	13.66	82.0	13.4	68.6	81.1				4.94	354	211	0.270	3.16
17J	5.13	12.83	81.3	12.9	68.4	80.0				5.05	324	209	0.268	3.13
18J	5.13	12.83	81.8	13.0	68.8	80.7				4.97	330	211	0.270	3.16
19J	6.64	16.60	81.2	11.0	70.2	79.9				5.08	418	343	0.439	5.14
20J	6.74	16.85	81.5	11.0	70.5	80.1				5.03	428	343	0.439	5.14
21J	6.82	17.05	82.0	18.2	63.8	81.0				6.78	322	196	0.251	2.93
22J	6.53	16.31	81.9	17.9	64.0	80.8				6.73	311	194	0.248	2.90
23J	6.58	16.47	81.5	13.8	67.7	80.2				5.81	363	266	0.340	3.98
24J	6.63	16.59	81.5	13.5	68.0	80.1				5.80	366	268	0.343	4.01
1GX	4.82	11.70	92.5	9.1	83.4	90.6				4.63	324	332	0.425	2.48
2GX	4.61	11.20	91.9	9.0	82.9	90.0				4.70	305	333	0.426	2.49
3GX	4.40	10.69	92.5	8.9	83.6	90.5				4.65	294	329	0.421	2.46

TABLE 1 (Concluded)

PRINCIPAL RESULTS OF TESTS ON HORIZONTAL SHELL-AND-TUBE CONDENSER

Test Number	Ammonia lb. per min.	Condenser Tonnage	Average Ammonia Saturation Temp. deg. F.	Diff. between Ammonia Saturation Temp. and Initial Water Temp. deg. F.	Initial Water Temp. deg. F.	Final Water Temp. deg. F.	Average Water Film Temp. deg. F.	Mean Temp. Drop through Ammonia Film deg. F.	Mean Temp. Drop through Tube Wall deg. F.	Mean Temp. Diff. between Ammonia and Water, deg. F.	Coefficient of Heat Transfer B.t.u. per hr. per sq. ft. per deg. F.	Condenser Water lb. per min.	Condenser Water gal. per min. per sq. ft.	Condenser Water ft. per sec.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
4GX	4.51	10.96	92.3	9.4	82.9	90.3				4.93	285	326	0.417	2.44
5GX	4.53	11.01	91.8	9.2	82.6	89.8				4.89	288	328	0.419	2.45
6GX	4.53	11.01	91.7	9.2	82.5	89.8				4.76	296	322	0.412	2.41
7GX	4.77	11.60	92.9	9.5	83.4	90.6				5.12	289	342	0.437	2.56
8GX	4.82	11.70	92.4	9.4	83.0	90.2				5.03	298	335	0.429	2.51
9GX	4.72	11.48	92.8	9.4	83.4	90.8				5.03	292	328	0.419	2.46
10GX	4.74	11.51	92.0	9.5	82.5	89.6				5.20	283	333	0.426	2.49
11GX	4.79	11.65	92.4	9.8	82.6	90.0				5.35	278	333	0.426	2.49
12GX	4.73	11.50	92.3	9.5	82.8	89.9				5.21	282	331	0.424	2.48
13GX	5.04	12.24	92.7	10.2	82.5	89.9				5.75	272	330	0.423	2.48
14GX	4.82	11.70	92.1	10.2	81.9	89.4				5.68	264	332	0.425	2.48
15GX	4.94	12.02	92.1	10.3	81.8	89.2				5.77	266	331	0.424	2.48
16GX	4.68	11.38	91.7	10.0	81.7	88.9				5.64	258	331	0.424	2.48
17GX	5.09	12.38	91.9	10.3	81.6	89.0				5.84	272	329	0.421	2.46
18GX	4.73	11.49	92.2	13.1	79.1	90.5				5.94	248	212	0.272	1.59
19GX	4.86	11.82	92.3	13.9	78.4	90.4				6.40	236	209	0.268	1.56
20GX	4.72	11.48	92.3	13.9	78.4	90.2				6.40	229	209	0.268	1.56
21GX	4.72	11.48	92.0	13.9	78.1	90.0				6.44	228	209	0.268	1.56
22GX	4.73	11.49	92.5	13.9	78.6	90.2				6.54	225	210	0.269	1.57
23GX	4.71	11.47	92.0	13.9	78.1	89.8				6.54	224	210	0.269	1.57
24GX	4.72	11.48	92.3	14.0	78.3	90.0				6.59	223	209	0.268	1.56
25GX	4.73	11.49	92.4	14.0	78.4	90.1				6.62	222	206	0.264	1.54
26GX	4.69	11.40	92.1	14.0	78.1	89.7				6.67	218	211	0.270	1.58
27GX	4.72	11.48	92.4	13.8	78.6	90.0				6.60	222	212	0.272	1.59
28GX	4.95	12.03	91.8	14.5	77.3	89.1				7.07	218	208	0.266	1.56
29GX	4.88	11.87	92.7	15.0	77.7	89.7				7.42	204	207	0.265	1.55
30GX	4.78	11.62	92.9	15.0	77.9	90.0				7.44	200	207	0.265	1.55
31GX	4.86	11.81	92.4	14.9	77.5	89.4				7.48	202	208	0.266	1.56
32GX	4.87	11.82	92.8	14.9	77.9	89.7				7.51	202	208	0.266	1.56
33GX	4.77	11.60	91.9	14.5	77.4	89.0				7.22	205	209	0.268	1.56

form the basis for the derivation of the performance charts in Section 15 as well as for all of the subsequent curves by means of which comparisons are made in regard to unit quantities, and afford a means of comparing total performances of the condenser with different amounts and arrangements of the condensing surface, and under different conditions of operation. The points on the curves represent the data as computed from the individual tests, and have been plotted against primary scales of ordinates consisting of the differences between the observed ammonia saturation temperatures and the observed initial temperatures of the cooling water, in order to eliminate the effect of the small deviations occurring in the separate tests from the liquefaction pressures of 145, 172, or 122 lb. per sq. in. gage predetermined for the respective series. The secondary scales designated as initial water temperatures were obtained by subtracting the differences shown on the primary scales from the fixed values of 81.7, 92, or 73.8 deg. F., or the saturation temperatures corresponding to the liquefaction pressures of 145, 172, and 122 lb. per sq. in. gage, respectively. The scale of initial water temperatures on each group of curves, therefore, represents the initial water temperature corrected to a common ammonia saturation temperature for that particular group. Since these corrections were always small, the curves may be regarded as representative of the actual conditions under which the tests were run.

11. *Effect of Liquefaction Pressure on Condenser Tonnage.*—The curves in Fig. 5 have been derived from those in Figs. 2 to 4 for a common initial water temperature of 68 deg. F., in order to compare the condenser tonnages developed under different operating conditions. The effect of liquefaction pressure may be observed by comparing the curves for the F, G, and H series. In these three series the water passed through the two shells of the condenser in parallel, and the F series, which may be regarded as the basic series, was run with a liquefaction pressure, or condenser pressure, of 145 lb. per sq. in. gage. This pressure was 172 lb. per sq. in. gage for the G series and 122 lb. per sq. in. for the H series. It may be noted that for a given water temperature and given weight of cooling water per min. the total tonnage developed increased materially as the liquefaction pressure was increased. The curve in Fig. 6, which was derived from Fig. 5 for a constant water rate of 150 lb. per min., indicates that the increase in total tonnage was directly proportional to the increase in liquefaction pressure. Since the initial water temperature of 68 deg. F. was selected, the tonnage developed would necessarily be zero at a

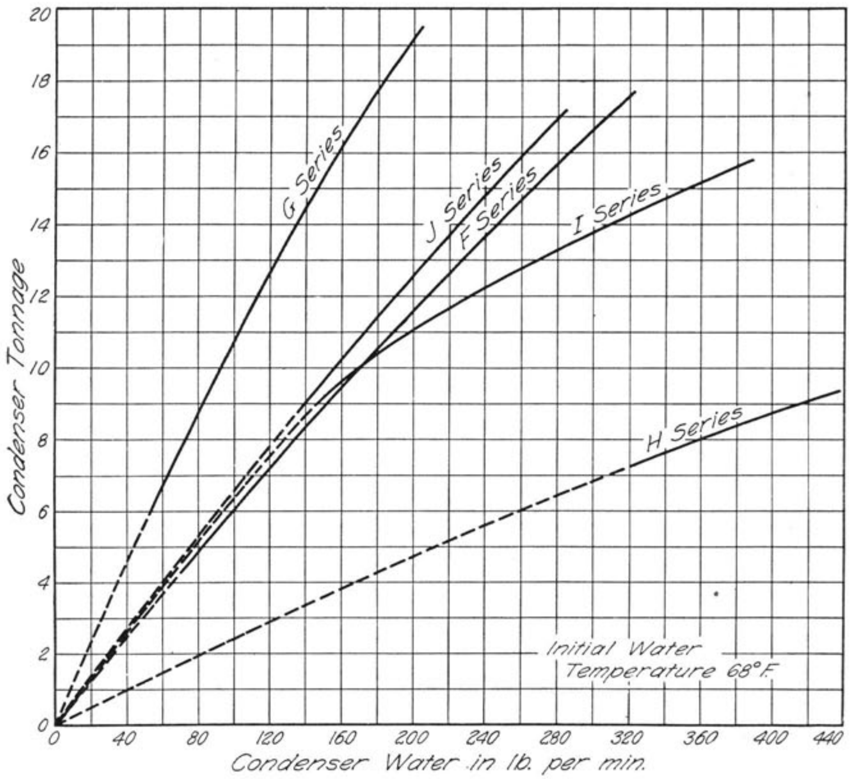

FIG. 5. TOTAL TONNAGE AND WATER RATE FOR F, G, H, I, AND J SERIES OF HORIZONTAL SHELL-AND-TUBE CONDENSER TESTS

liquefaction pressure of 109.5 lb. per sq. in. gage, corresponding to an ammonia saturation temperature of 68 deg. F.

The increase in total tonnage with an increase in liquefaction pressure, at a given initial water temperature and a given water rate, results from the fact that the tonnage developed is a function of the mean temperature difference between the ammonia vapor and the water, and as the saturation temperature corresponding to the liquefaction pressure is increased the mean temperature difference is also increased. Hence, it is obvious that if a condenser is small, or if the temperature of the available cooling water is high, the capacity may be increased by operating at a higher head pressure. This, of course, would involve more work done in compression, and might involve certain adaptations in the compressor itself. The actual total capacity, or rating, of a condenser is governed by the amount of effec-

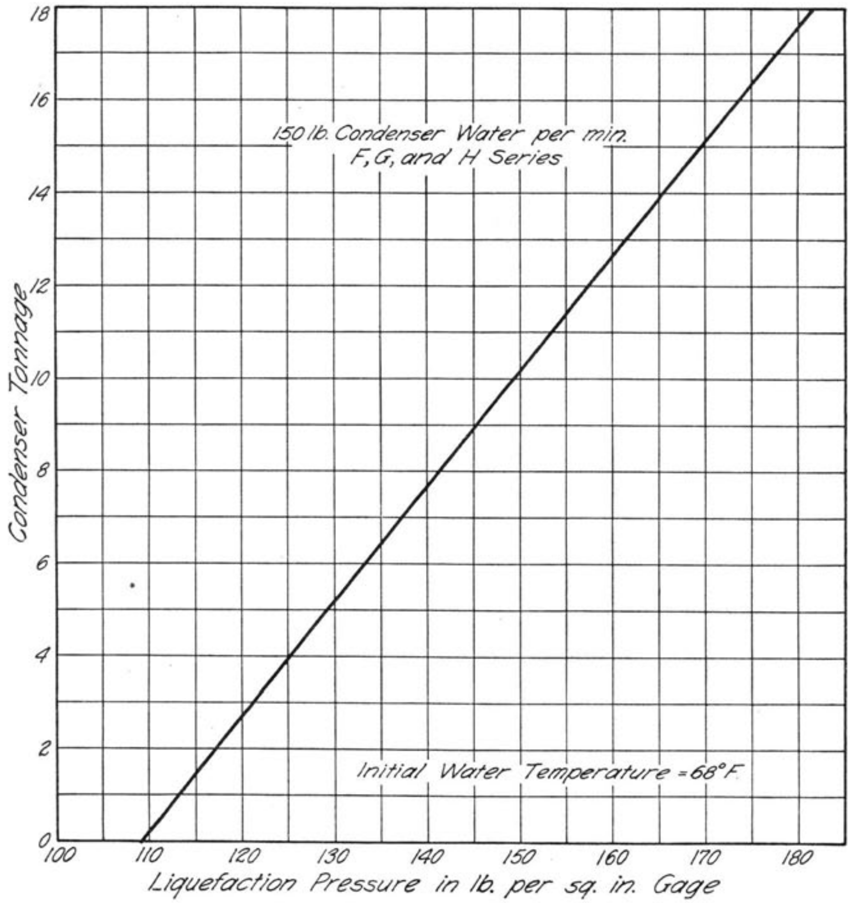

FIG. 6. TOTAL TONNAGE AND LIQUEFACTION PRESSURE FOR F, G, AND H SERIES OF HORIZONTAL SHELL-AND-TUBE CONDENSER TESTS

tive condensing surface, the limiting condenser pressure, the temperature of the water available, and the amount of water circulated.

The curves in Figs. 5 and 6 indicate that for a given initial water temperature and water rate the ammonia saturation temperature determines the mean temperature difference, or temperature head, and consequently the total capacity of the condenser, but give no indication as to whether the rate of condensation or rate of heat transfer is influenced by the saturation temperature itself, apart from its influence as a factor determining the temperature head. The curve in Fig. 7 was therefore plotted, giving the relation between both

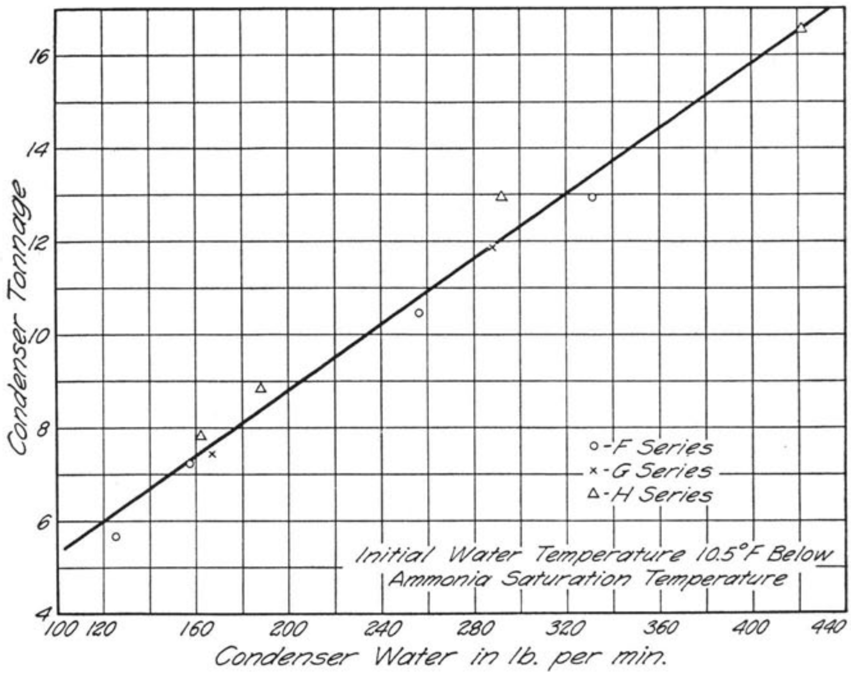

FIG. 7. TOTAL TONNAGE FOR DIFFERENT WATER RATES WITH CONSTANT DIFFERENCE IN TEMPERATURE BETWEEN SATURATED AMMONIA AND INITIAL WATER FOR F, G, AND H SERIES OF HORIZONTAL SHELL-AND-TUBE CONDENSER TESTS

water rate and condenser tonnage for a common temperature head, or difference in temperature between ammonia saturation temperature and initial water temperature, of 10.5 deg. F. for the three series. The distribution of points for the three series indicates that, for a given temperature head, the liquefaction pressure had no material influence on the condenser tonnage developed for a given water rate in pounds per minute. The curve further indicates that, over the range of the tests, for a given temperature head, the increase in tonnage developed was directly proportional to the increase in the pounds of cooling water used per minute.

12. *Effect of Changes in Condensing Surface on Condenser Tonnage.*—The effect of changing the arrangement and amount of condensing surface may be observed by comparing the curves for the F, I, and J series in Fig. 5. For these three series the liquefaction pressure was maintained at 145 lb. per sq. in. gage. For the F series both shells of the condenser having a total condensing surface of 93.8 sq. ft.

were used, and the water passed through the two shells in parallel. For the J series the water was passed through the two shells in series, and the total condensing surface was still 93.8 sq. ft. For the I series but one shell was used, and the condensing surface was 46.9 sq. ft.

From the curves in Fig. 5 it may be noted that, for a given initial water temperature, and with a given weight of cooling water per minute, a greater tonnage was developed by the J series than by the F series. When the water was passed through the two shells in series the effect was to double the velocity of the water through the tubes for a given weight of water used per minute, as compared with the velocity obtained when the two shells were used in parallel. While both the change in arrangement and the change in water velocity were accompanied by readjustments in the rate of heat transfer, as explained in Section 13, the effect of increased water velocity predominated, and the net result was an increase in total tonnage developed.

When one shell alone was used, the velocity of the water passing through the tubes was also doubled as compared with the velocity obtained when the same weight of water per minute was passed through the two shells in parallel. The curve for the I series, shown in Fig. 5, indicates that up to a limit of approximately 10 tons the increased rate of heat transfer resulting from the greatly increased water velocity was more than sufficient to offset the effect of the reduced amount of surface and the tonnage developed probably was, if anything, slightly greater than that developed by the F series. Above this limit, the reduction in condensing surface more than offset the effect of increased water velocity, and resulted in a rapid reduction in capacity as compared with the capacity developed by the F series. The curve for the I series does not belong to the same family of curves as those for the other series, and indicates that with only one shell in operation the condenser became overloaded at rates above 10 tons. With both shells in operation no overloading became apparent over the range of tests run.

13. *Coefficients of Heat Transfer.*—The effects of both water velocity and arrangement of condensing surfaces on the coefficients of heat transfer are shown in Fig. 8. The rate of heat transfer increased rapidly as the mean velocity of the water passing through the tubes was increased.

The arrangement of condensing surface was essentially the same for the F, G, H, and I series. The length of water travel with respect to the condensing surface was the same in the case of the I series,

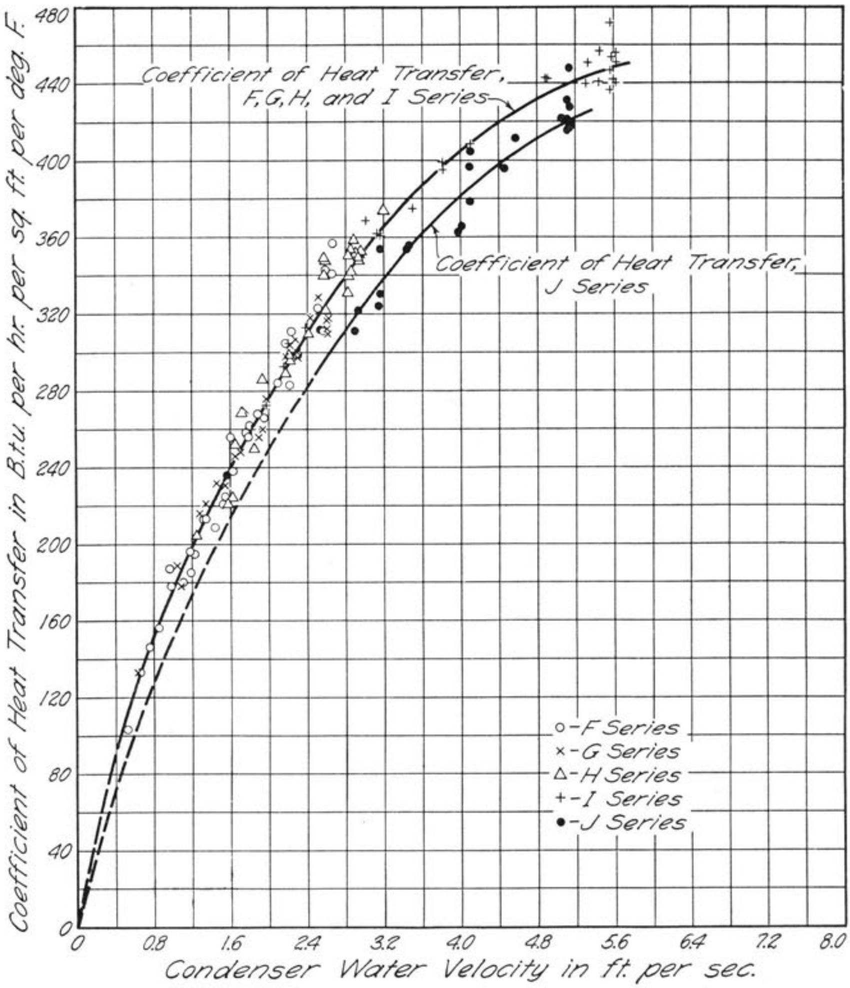

FIG. 8. COEFFICIENTS OF HEAT TRANSFER FOR DIFFERENT WATER VELOCITIES FOR F, G, H, I, AND J SERIES OF HORIZONTAL SHELL-AND-TUBE CONDENSER TESTS

where only one shell was used, as it was in the cases of the F, G, and H series, where the two shells were used in parallel, since the two shells were duplicates. Furthermore, in the latter case, the initial water temperature was the same for each shell. In order to obtain the same water velocity in each shell in the parallel arrangement, the total weight of water used for the condenser as a whole would have to be twice that used for a single shell. Under these conditions, the two shells would act as two separate and distinct condensers, and the capacity of the condenser as a whole would be twice the capacity developed by a single shell. Hence it is evident that, when compared on the basis of the same water velocity, the single and double shell condensers would exhibit the same characteristics in regard to the rate of heat transfer, and the points representing the coefficients of heat transfer would all fall on the same curve, as shown for the F, G, H, and I series in Fig. 8.

The arrangement of condensing surface for the J series, in which the two shells were used in series, was fundamentally different from that used in the F, G, H, and I series in that the length of water travel with respect to the condensing surface was twice as great. With the same initial water temperature and the same water velocity a difference in the mean temperature difference between the ammonia vapor and the water, and accordingly in the heat transfer characteristics, would be expected in the two cases. Figure 8 indicates that under these conditions the greater length of water travel resulted in a greater mean temperature difference, and consequently in smaller values for the mean coefficients of heat transfer, as shown by the curve for the J series. It is further indicated that the most effective arrangement of condensing surface is one in which the water travel is comparatively short, and which offers a large area for the action of the coldest water near the point of entry.

The curves in Fig. 9 afford a comparison of the coefficients of heat transfer on the basis of water circulated per minute per square foot of surface. For a given volume of cooling water circulated per minute per square foot the water velocity would be the same for the single shell and for the two shells arranged in parallel. Hence the results for the F, G, H, and I series are again all represented by the same curve.

In the case of the J series, in which the two shells were used in series, a given volume of water circulated per minute per square foot represents twice the water velocity obtained when the same volume of water was circulated per minute per square foot in the parallel or single shell arrangements used for the F, G, H, and I series. Hence, while the effect of the arrangement of surfaces alone was such as to reduce the coefficient of heat transfer as shown in Fig. 8, the increase resulting from increased water velocity for a given volume of water circulated per minute per square foot was more than enough to offset the reduction due to arrangement of surface alone, and the net result, as shown in Fig. 9, was to materially increase the values of the coefficients of heat transfer for the J series as compared with those for the F, G, H, and I series for the same unit volume of water circulated.

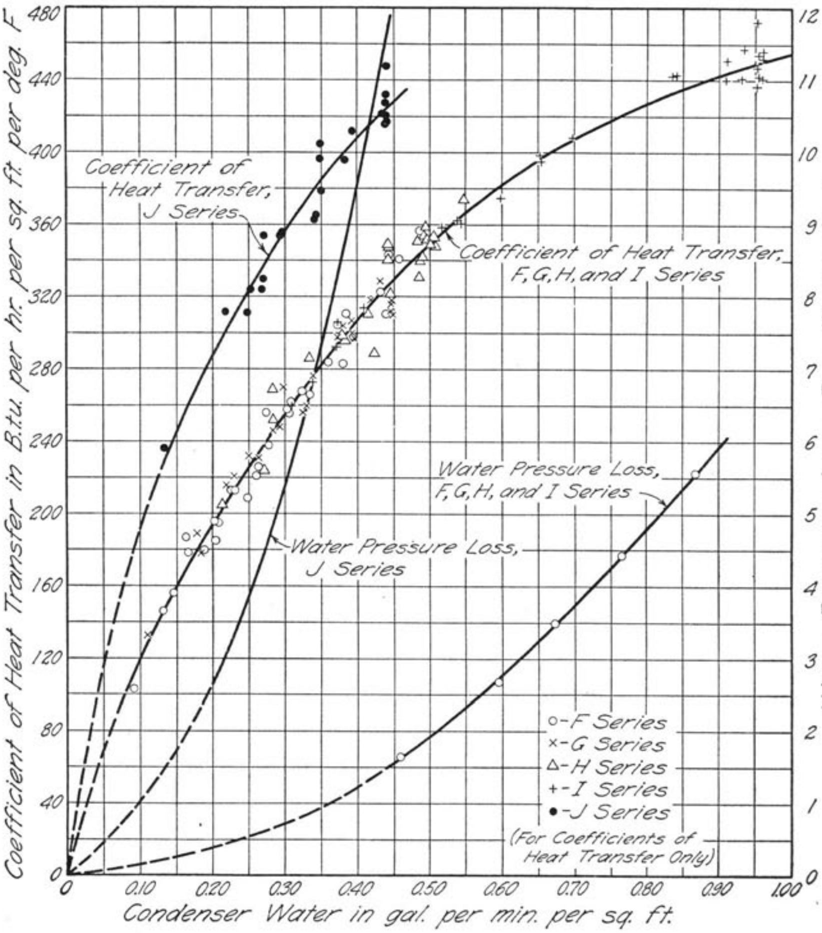

FIG. 9. COEFFICIENTS OF HEAT TRANSEFR AND FRICTION PRESSURE LOSS FOR DIFFERENT UNIT WATER RATES FOR F, G, H, I, AND J SERIES OF HORIZONTAL SHELL-AND-TUBE CONDENSER TESTS

The curves in Fig. 9 serve also further to explain the apparently anomalous curve for the I series in Fig. 5, since for a given total weight of water circulated per minute the unit volume circulated per minute per square foot becomes comparable with that represented by the J series in Fig. 9, and the coefficient of heat transfer would be nearly the same as that shown for the J series in this figure. Hence, up to the limit at which the smaller amount of surface represented by the I series in Fig. 5 became overloaded it is probable that the capacity

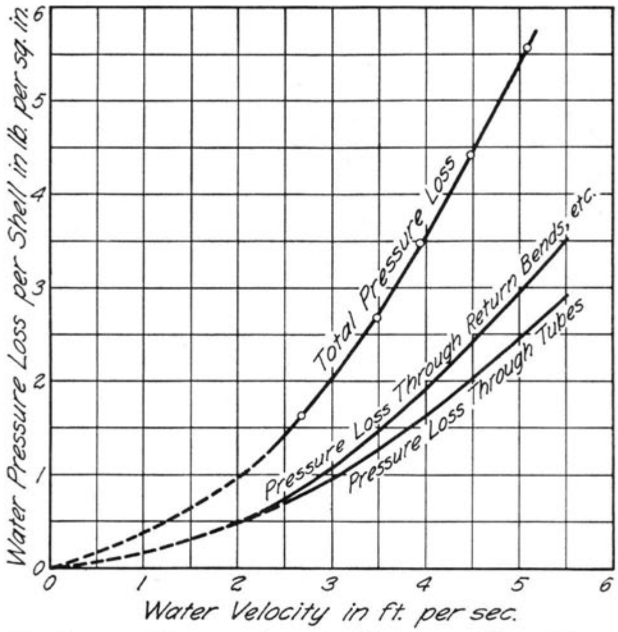

FIG. 10. FRICTION PRESSURE LOSS FOR DIFFERENT WATER VELOCITIES FOR HORIZONTAL SHELL-AND-TUBE CONDENSER TESTS

developed by the single shell would be nearly as great as that developed by the two shells in series.

14. *Pressure Loss Through Condenser Tubes.*—The curves for total friction pressure loss, loss through the return bends, and through the tubes alone, are shown in Fig. 10. These data were obtained for a water temperature of 60 deg. F. by means of mercury manometers placed at the entrance and exit of one shell of the condenser.

The friction pressure loss through the straight tubes alone under various conditions of water velocity and temperature may readily be calculated by making use of the Reynolds' number, as outlined in Bulletin No. 182.* By subtracting this calculated loss from the total measured friction pressure loss, shown by the upper curve in Fig. 10, an approximation of the loss resulting from entry conditions, return bends, and changes in cross-sectional area may be obtained. Both the loss through the tube alone and the loss in the return bends, etc., are shown in Fig. 10, and it may be noted that the loss through the return bends forms a relatively large proportion of the total friction loss.

*"Flow of Brine in Pipes," Univ. of Ill. Eng. Exp. Sta. Bul. No. 182, 1928.

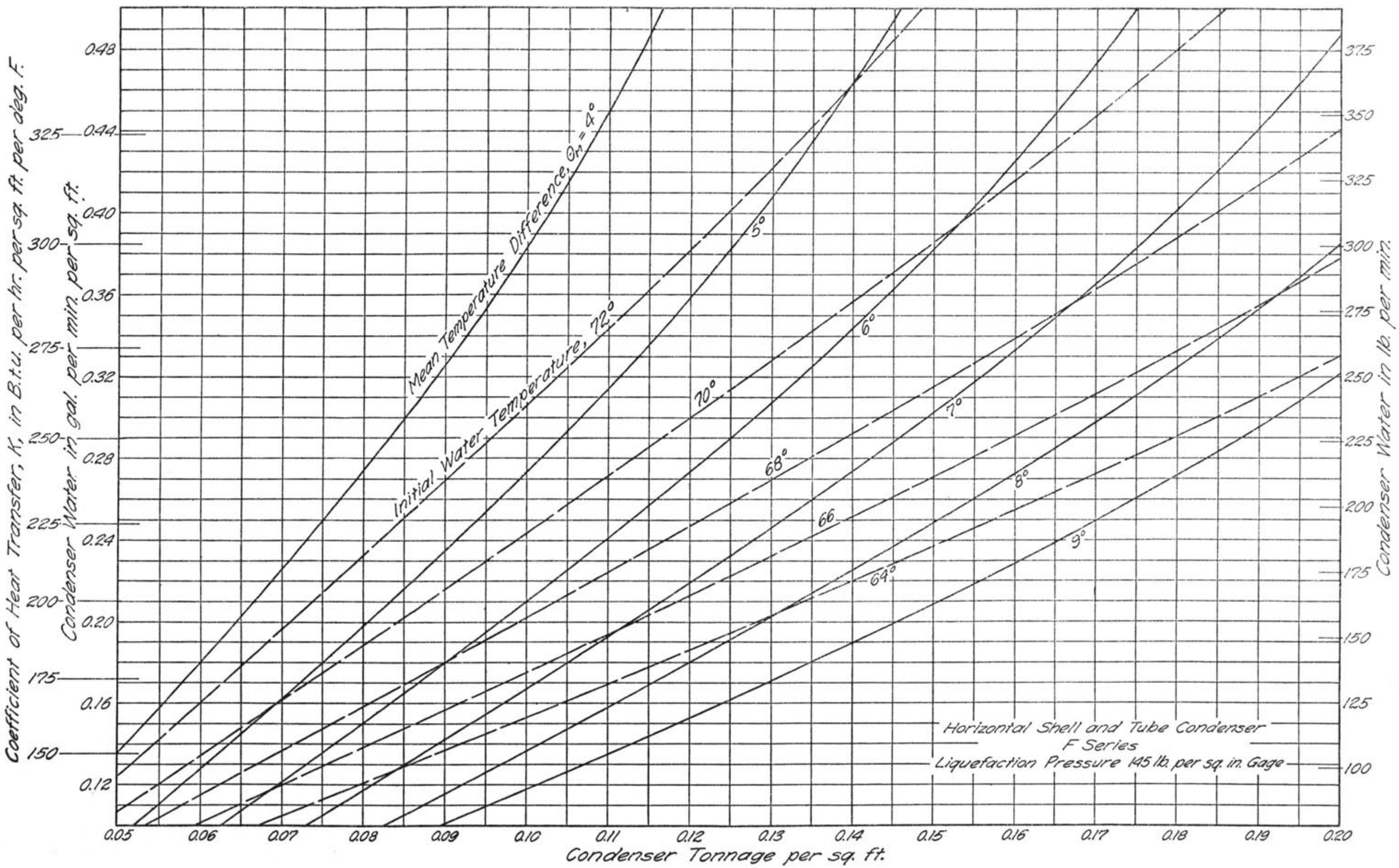

FIG. 11. PERFORMANCE CHART FOR F SERIES OF HORIZONTAL SHELL-AND-TUBE CONDENSER TESTS

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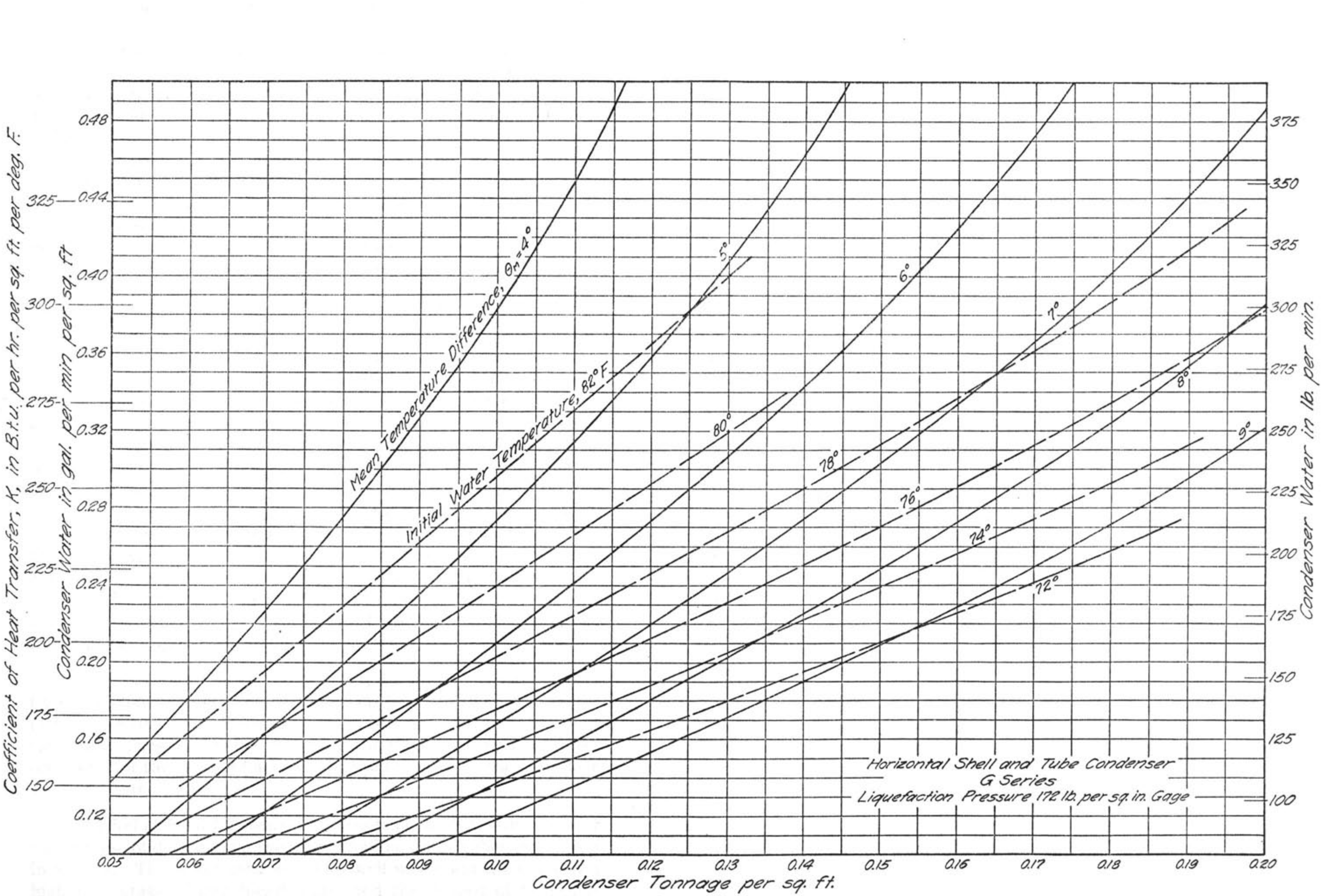

FIG. 12. PERFORMANCE CHART FOR G SERIES OF HORIZONTAL SHELL-AND-TUBE CONDENSER TESTS

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In order to compare the friction pressure losses occurring under the conditions of service established for the different series of tests with the coefficients of heat transfer obtained under these same conditions, the curves for total friction pressure loss have also been shown in Fig. 9. Since for a given volume of water circulated per minute per square foot of surface the water velocity would be the same for the two shells in parallel as for the single shell, the pressure loss per shell would also be the same. Also, in the case of the parallel arrangement, the total pressure required to circulate the water through the condenser as a whole would be the same as the friction pressure loss per shell. Therefore, when compared on the basis of water circulated per minute per square foot of surface the same curve would represent the total friction pressure loss for the F, G, H, and I series.

In the case of the J series, since a given volume of water circulated per minute per square foot of surface represents twice the water velocity obtained for the F, G, H, and I series under the same conditions, the friction pressure loss per shell would be four times as great. Furthermore, since the two shells were used in series, the total pressure required to circulate the water would be twice the friction pressure loss per shell. Hence, for a given volume of water circulated per minute per square foot of surface, the total friction pressure loss for the J series, as shown in Fig. 9, was eight times the corresponding friction pressure loss for the F, G, H, and I series. Thus it is evident that, although the coefficient of heat transfer and total capacity of the condenser for a given amount of water circulated may be materially increased by connecting the two shells in series instead of in parallel, this increase is necessarily accompanied by a corresponding increase in the power required to circulate the water.

15. *Performance Charts.*—Figures 11, 12, and 13 consist of performance charts representing the relations between all of the various factors affecting the performance of the condenser over the range of the tests. These charts are shown for the three condenser pressures used, and for the standard arrangement only, in which the two shells were connected in parallel.

16. *Effect of Fouled Tubes.*—The results of the GX series, run for the purpose of determining the rate of fouling of the tubes, are shown in Fig. 14. The water used was deep well water which contained a high percentage of iron bicarbonate, and the deposit of ferric hydroxide that formed was more of the nature of a soft sludge than of a true scale. The effect of this sludge was to cause a gradual decrease

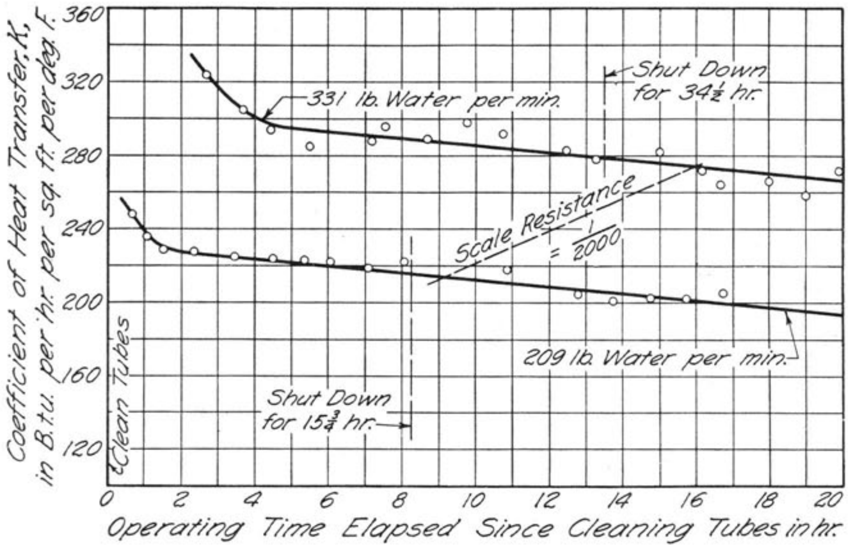

FIG. 14. RATE OF DECREASE IN COEFFICIENT OF HEAT TRANSFER WITH FORMATION OF SCALE IN TUBE OF HORIZONTAL SHELL-AND-TUBE CONDENSER

in the coefficient of heat transfer and a corresponding decrease in the temperature of the water required to maintain a constant tonnage with a given water rate and condenser pressure.

At each of the water velocities used the first layers of scale formed during the first two to four hours of the run had the most marked effect on the heat transfer, as indicated by the rapid decrease in the coefficient of heat transfer shown in Fig. 14. After approximately five hours of running the effect of the scale formation became very much less, and also became practically the same for the two water velocities.

It was not found possible to obtain consistent results when tests were run during the first four hours after the tubes had been cleaned. The effect of the formation of foreign film during the first four hours of running was regarded as probably being characteristic, independent of the nature of the water used, and the condition of the tubes at this time was considered as commercially clean. The tests for all series were, therefore, started at the end of a four-hour period, and testing was never continued beyond six hours from this time before the tubes were again cleaned.

Test XF was run with excessively fouled tubes, and the difference between the ammonia saturation temperature and that of the water

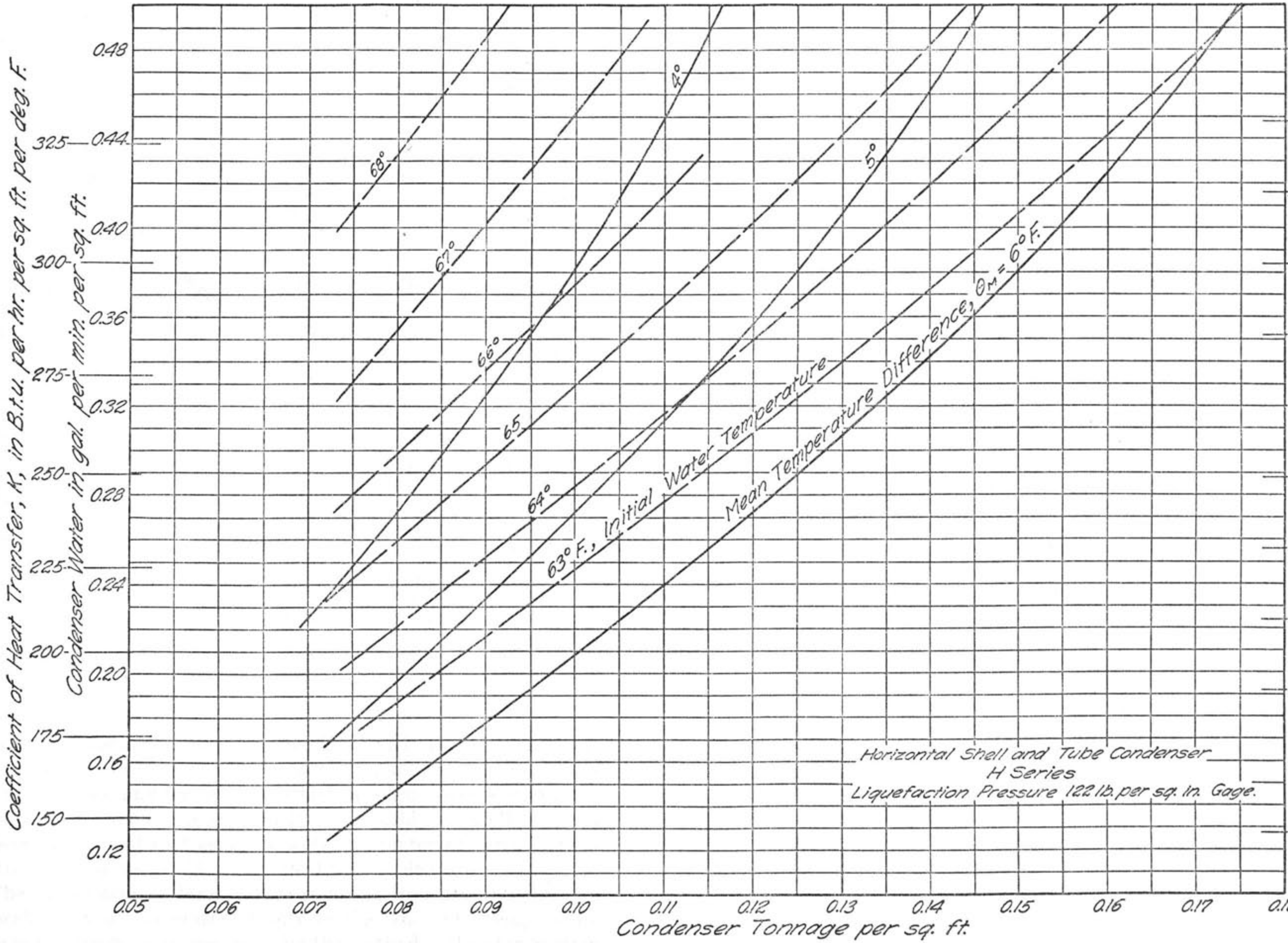

FIG. 13. PERFORMANCE CHART FOR H SERIES OF HORIZONTAL SHELL-AND-TUBE CONDENSER TESTS

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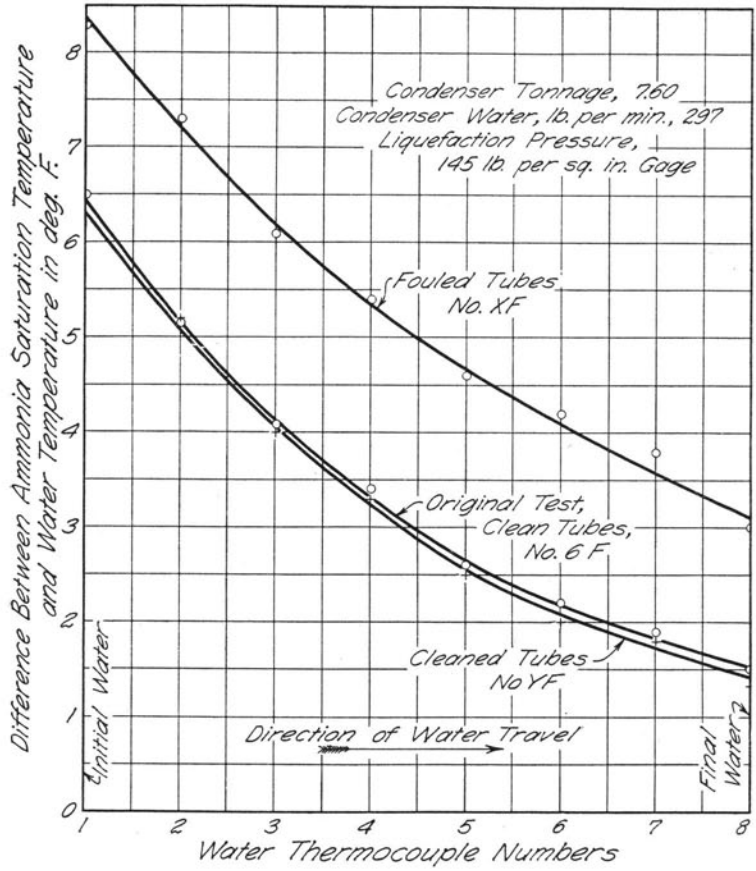

FIG. 15. EFFECT OF SCALE ON DIFFERENCE BETWEEN AMMONIA SATURATION TEMPERATURE AND WATER TEMPERATURE IN HORIZONTAL SHELL-AND-TUBE CONDENSER

for successive distances along the tubes is shown in Fig. 15. The same data are shown for test 6F, which was run under the same conditions with clean tubes, and for test YF which was run after cleaning the tubes. These curves indicate that it was possible to duplicate tests on clean tubes, and that erratic results and materially greater temperature differences would be obtained with fouled tubes.

The added drop in temperature resulting from the resistance of the scale is shown diagrammatically in Fig. 16. The temperature drops in this figure were computed by the method outlined in Section 21, and it is evident that the additional temperature drop through the scale in the case of test XF was approximately 50 per cent of the total

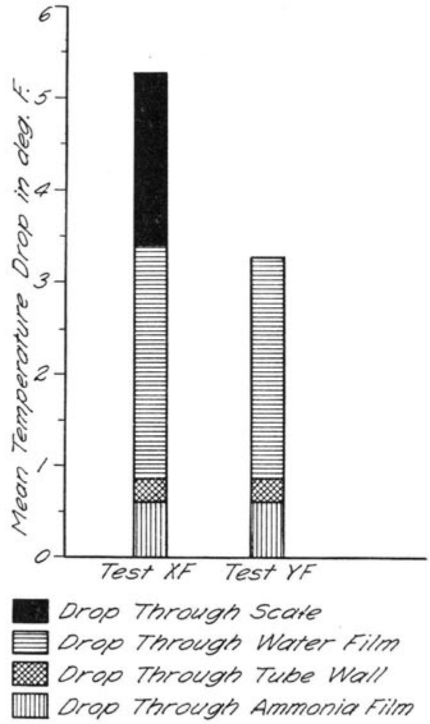

FIG. 16. TEMPERATURE GRADIENT THROUGH TUBES OF HORIZONTAL SHELL-AND-TUBE CONDENSER

temperature drop from the saturated ammonia to the water through the clean tube in test YF. Further discussion of the coefficients of heat transfer as affected by fouled tubes is given in Section 22.

17. *Comparison of Performance of Four Types of Condensers.*—The investigation of heat transfer in ammonia condensers has included four different types of condensers, namely, the atmospheric-bleeder, the double-pipe, the vertical shell-and-tube, and the horizontal shell-and-tube or multitube-multipass. Taking into consideration modifications made in the arrangement of condensing surfaces for purposes of special tests, the total amount of condensing surface has varied from 92 to 251 sq. ft. Under given conditions, the total tonnage developed has been determined largely by the total condensing surface, and comparisons involving total tonnage are therefore of practically no value. Such comparisons must be made on the basis of

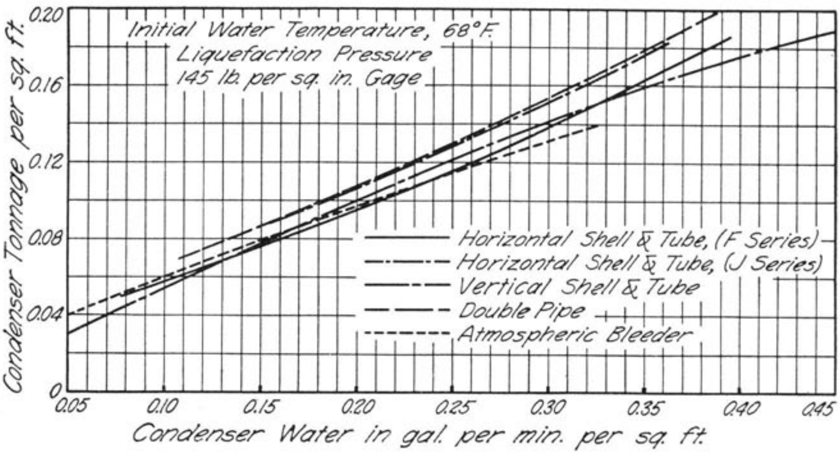

FIG. 17. COMPARISON OF PERFORMANCE OF FOUR TYPES OF CONDENSERS

unit values, or of tonnage per square foot and water circulated per minute per square foot of condensing surface.

In order to compare the performance of the four types of condensers on a unit basis the curves for the multitube-multipass condenser have been superimposed on similar curves for the other types, which have been transferred from Bulletins Nos. 171 and 186. These curves for an initial water temperature of 68 deg. F. are shown in Fig. 17, and it is evident that, including water rates up to 0.31 gal. per min. per sq. ft., they all lie within a comparatively narrow band, the maximum width of which does not exceed 16 per cent of the mean value of the tonnage per square foot for the given unit water rate. It is therefore possible to represent the mean tonnage for all of the condensers tested by a single line. The actual tonnage per square foot developed by any one of the condensers did not deviate by more than 8 per cent from the value read from this line at a corresponding unit water rate. This procedure was also found valid for initial water temperatures other than 68 deg. F. The mean curves so determined for four different initial water temperatures are shown in Fig. 18.

The curves in Fig. 18 apparently indicate that, in so far as the condensers tested are concerned, the tonnage per square foot for a given amount of water circulated per minute per square foot is a figure of more practical value than the mean coefficient of heat transfer, since the latter varies widely with different conditions and with different arrangements of condensing surface, while the former can be represented by a set of mean curves, the deviations from which do

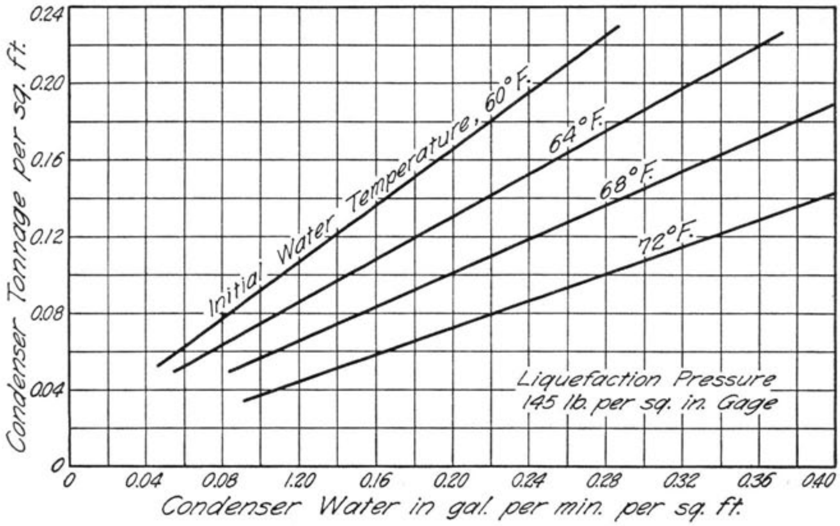

FIG. 18. UNIT TONNAGES AND UNIT WATER RATES FOR DIFFERENT INITIAL WATER TEMPERATURES FOR FOUR TYPES OF CONDENSERS

not exceed approximately 8 per cent over the range of conditions under which the tests were run.

V. APPLICATION OF GENERAL EQUATIONS FOR HEAT FLOW

18. *Flow Through Resistances in Series.*—By direct application of the principles involved in the film concept of heat flow through several resistances in series, the equation for the flow of heat through the resistance offered by the ammonia film, the tube, and the water film, in the case of a condenser with clean tubes, may be written

$$\frac{1}{KA_o} = \frac{1}{C_aA_o} + \frac{x_t}{k_tA_m} + \frac{1}{C_wA_i} \tag{1}$$

in which K = overall coefficient of heat transmission, based on area of condensing surface, B.t.u. per sq. ft. per degree temperature difference between the ammonia vapor and the water per hour.

k_t = conductivity of the metal tubes, B.t.u. per sq. ft. per degree temperature difference between the outside and inside surfaces of the tubes per ft. of tube thickness per hour.

C_a = conductance of the ammonia film, based on the area of the film, or the *outside* area of the tubes, and the actual thickness of film, B.t.u. per sq. ft. per degree temperature difference between the ammonia vapor and the outside surface of the tube per hour.

C_w = conductance of the water film, based on the area of the film, or the *inside* area of the tubes, and the actual thickness of film, B.t.u. per sq. ft. per degree temperature difference between the inside surface of the tube and the water per hour.

A_o = area of the outside surface of the tubes, sq. ft.

A_i = area of the inside surface of the tubes, sq. ft.

A_m = mean of the areas of the outside and inside surfaces of the tubes, sq. ft.

x_t = thickness of tube wall, in ft.

Equation (1) may be written in terms of the area of the condensing surface as follows:

$$\frac{1}{KA_o} = \frac{1}{C_aA_o} + \frac{x_t}{k_tA_o}\left(\frac{2d_o}{d_i + d_o}\right) + \frac{1}{C_wA_o}\left(\frac{d_o}{d_i}\right) \tag{2}$$

in which d_o = outside diameter of tubes, ft.
d_i = inside diameter of tubes, ft.

From Equation (2)

$$\frac{1}{K} = \frac{1}{C_a} + \frac{x_t}{k_t}\left(\frac{2d_o}{d_i + d_o}\right) + \frac{1}{C_w}\left(\frac{d_o}{d_i}\right) \tag{3}$$

The terms in Equation (3) now represent the total resistance to heat flow, the resistance of the ammonia film, the resistance of the clean tubes, and the resistance of the water film, respectively, all expressed as functions of the area of the condensing surface.

Since it is customary to use condensing surface, or the surface in contact with ammonia vapor, as the reference surface in making all condenser computations, it will be convenient to use a virtual conductance for the water film C'_w, which may be defined as the conductance of the water film, based on condensing surface in B.t.u. per sq. ft. of condensing surface per degree temperature difference between the inside surface of the tubes and the water per hour. Then,

$$C'_wd_o = C_wd_i \tag{4}$$

$$C'_w = C_w \frac{d_i}{d_o} \tag{5}$$

Equation (3) then becomes

$$\frac{1}{K} = \frac{1}{C_a} + \frac{x_t}{k_t}\left(\frac{2d_o}{d_i + d_o}\right) + \frac{1}{C'_w} \tag{6}$$

From Equation (6) it is evident that if the resistances of the tubes and ammonia film are assumed to be constant, and if the resistance of the water film $\frac{1}{C'_w}$ is some function $\frac{1}{f(v)}$ of the water velocity v, a plot of $\frac{1}{K}$ as ordinates against $\frac{1}{f(v)}$ as abscissas will give a straight line. This form of plot was proposed by Wilson,* and has since been used by Haslam, Ryan, and Weber,† and by McAdams, Sherwood, and Turner‡ in evaluating the resistance of liquid films. These investigators have shown that for turbulent flow inside of tubes

$$\frac{1}{C_w} = \frac{1}{av^{0.8}} \tag{7}$$

in which a is a constant depending on the dimensions of the apparatus used.

Substituting the value of C_w from Equation (5) and combining $\frac{a\,d_i}{d_o}$ into a single constant b

$$\frac{1}{C'_w} = \frac{1}{b\,v^{0.8}} \tag{8}$$

in which b is also a constant depending on the dimensions of the apparatus used, and represents the slope of the line obtained by plotting the observed values of $\frac{1}{K}$.

A plot of the observed values of $\frac{1}{K}$ against the observed values of $\frac{1}{v^{0.8}}$ obtained from the four series of tests, F, G, H, and I, is shown in

*E. E. Wilson, Trans. A.S.M.E., Vol. 37, 1915, p. 47.
†R. T. Haslam, W. P. Ryan, and H. C. Weber, Industrial and Engineering Chemistry, Vol. 15, 1923, p. 1105.
‡W. H. McAdams, T. K. Sherwood, and R. F. Turner, Trans. A.S.M.E., Vol. 48, 1926, p. 1233.

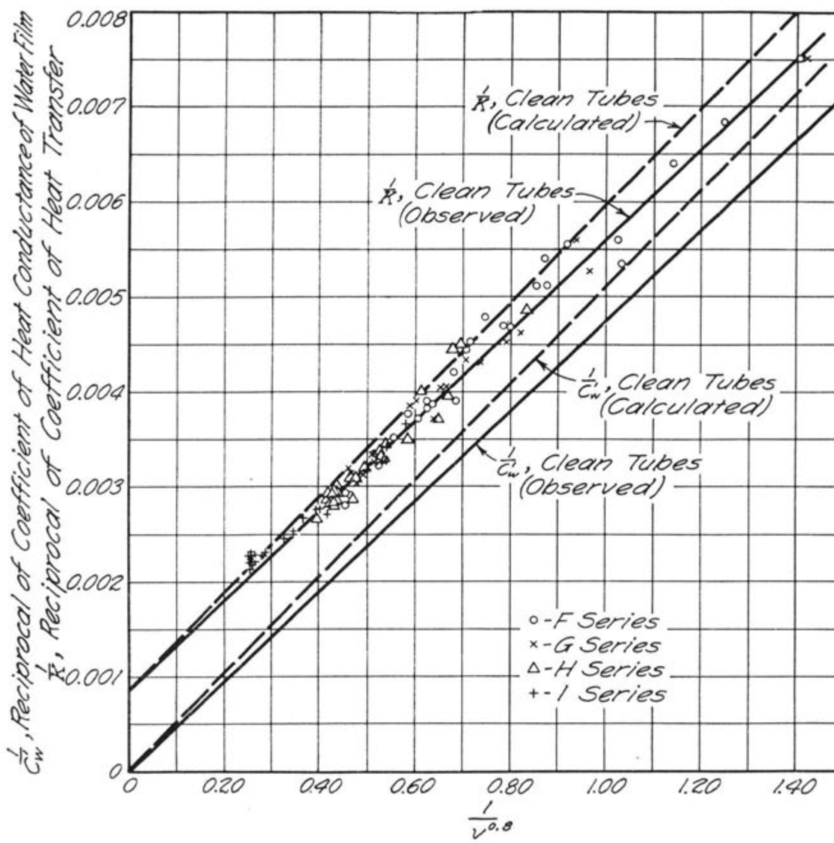

FIG. 19. OBSERVED AND CALCULATED COEFFICIENTS OF HEAT TRANSFER FOR CLEAN TUBES

Fig. 19. The points for the J series practically coincide with the ones shown, but have been omitted in order to avoid confusion. It was found by trial that the individual points representing the data deviated least from a straight line when the exponent 0.8 was used for v. Hence the performance of the multitube-multipass condenser is consistent with the performance of the apparatus used by the investigators cited, in that the resistance of the water film varies as the reciprocal of the 0.8 power of the water velocity.

Inspection of Equation (6) indicates that when $\frac{1}{C'_w}$ is zero, the intercept $\frac{1}{K}$ is the sum of the resistances of the ammonia film and of the clean tube. Furthermore, if these two resistances are assumed to remain constant over the range of the tests, a line drawn through the origin parallel to the line representing $\frac{1}{K}$ for clean tubes will represent the resistance of the water film $\frac{1}{C'_w}$, based on a unit of condensing surface.

19. *Resistance of Ammonia Film.*—If the conductivity of the metal in the tube is known, the conductance of the ammonia film may be calculated by making $\frac{1}{C'_w}$ zero, and substituting the value of the intercept for $\frac{1}{K}$ in Equation (6). In the case of the tests under discussion, the intercept, from Fig. 19, is 0.00085, the outside diameter of the tubes was 0.1667 ft., the inside diameter 0.1508 ft., and the thickness of tubes 0.00795 ft. The tubes were made of charcoal iron and the conductivity k_t was taken as 34.9.* Substituting in Equation (6)

$$0.00085 = \frac{1}{C_a} + \frac{0.00795 \times 2 \times 0.1667}{34.9 \times (0.1667 + 0.1508)} + 0$$

$$C_a = 1635.$$

The value, 1635, thus found is somewhat higher than the value 952 deduced by Sherwood† for vertical shell and tube condensers. Sherwood based his computations on a conductivity of 25 for the metal tube. If this value were used in the present computations a value of 1940 for C_a would result. In both cases the effect of scale or rust on the tube has been neglected. If the scale were taken into consideration the effect would be to increase the calculated value of C_a. The presence of a greater amount of scale in the condensers investigated by Sherwood than in the one used on the tests under discussion might serve as a possible explanation of the difference in the values of C_a observed, since the greater scale resistance would cause a proportionally larger reduction in the value of C_a for the vertical shell and tube condenser. However, it is not probable that this would account for so large a difference. The tests on the vertical shell and tube condensers included in Sherwood's analysis were run under somewhat different conditions than the ones on the multitube-multipass con-

*Mark's Handbook, p. 303.
†T. K. Sherwood, Refrigerating Engineering, Vol. 13, No. 8, February, 1927, p. 253.

denser, in that in the former the superheat was removed in the condenser itself, while in the latter a separate superheat remover was used. Hence, the average conductance determined by Sherwood included that of the gas film in the superheated region. The conductance of the superheated gas film would be lower than that for saturated vapor or liquid film, and the average conductance would be lower than the average determined under conditions where the whole condenser contained saturated vapor only. Therefore, it seems reasonable that the value 1635 is more representative of the conductance of the saturated ammonia film than the value 952.

20. *Resistance of Water Film.*—The resistance of the water film per square foot of condensing surface is shown in Fig. 19 by the solid line, $\frac{1}{C'_w}$, which passes through the origin. The conductance of the water film per square foot of condensing surface may be calculated from this line, or from the empirical equation representing it.

$$C'_w = 211\ v^{0.8} \tag{9}$$

The idea that heat is transmitted from a flowing fluid to a surface by conduction through a stationary film of the fluid at the surface, and that the thickness of this film, and hence the rate of heat conduction through it, is some function of the frictional resistance at the surface, probably originated with Osborne Reynolds. This theory has been extended by Major C. I. Taylor,* T. E. Stanton, J. R. Pannell, and a number of other investigators.

Since the frictional resistance is a function of linear velocity, density, and viscosity of the liquid, and inside diameter of the tube, and $C_w = \frac{k_w}{x_w}$, Rice,† and McAdams and Frost‡ have shown by the principle of dimensional analysis that

$$C_w = \frac{k_w}{d_i}\phi\left(\frac{d_i v\rho}{\mu}\right)^n \tag{10}$$

in which C_w = conductance of a water film of thickness x_w, B.t.u. per hr. per deg. F. per sq. ft. of water surface of the tube.

k_w = conductivity of water, B.t.u. per hr. per sq. ft. per deg. F. per ft. of thickness.

*British Advisory Committee for Aeronautics, Reports and Memoranda, Nos. 243 and 272.
†C. W. Rice, Indus. & Eng. Chem., Vol. 16, No. 5, May, 1924, p. 460.
‡W. H. McAdams and T. H. Frost, Indus. & Eng. Chem., Vol. 14, No. 12, December, 1922, p. 1101.

d_i = inside diameter of tube, ft.
v = velocity of water, ft. per sec.
ρ = density of water, lb. per cu. ft.
μ = absolute viscosity of water at the mean temperature of the film, lb. per ft. per sec.

By making use of these principles, and introducing a factor to correct for entrance conditions, McAdams and Frost* have deduced the equation

$$C_w = \frac{k_w}{d_i}\left(1 + \frac{N}{r}\right)\phi\left(\frac{d_i v \rho}{\mu}\right)^n \tag{11}$$

in which N = number of inside tube diameters to be added to the actual length of tube expressed in diameters, to obtain the effective length for heat transfer.
r = ratio of actual tube length to actual inside diameter.

The ratio of the absolute viscosity to the density $\frac{\mu}{\rho}$ is defined as the kinematic viscosity. By letting ν = the kinematic viscosity in sq. ft. per sec., Equation (11) may be written

$$C_w = \frac{k_w}{d_i}\left(1 + \frac{N}{r}\right)\phi\left(\frac{d_i\, v}{\nu}\right)^n \tag{12}$$

In order to evaluate N and to determine the nature of the function $\frac{d_i\, v}{\nu}$ McAdams and Frost, using the data of Stanton,† Webster,‡ and Bray and Sayler,§ plotted $\frac{C_w d_i}{k_w}$ against $\frac{d_i v}{\nu}$. These curves have been replotted to conform to the units used in this paper, and are shown in Fig. 20. Since the average slope of the straight lines shown is 0.8, it is evident that $\frac{d_i v}{\nu}$ in Equation (12) must also have the exponent 0.8, or

$$\frac{C_w d_i}{k_w} = \phi\left(1 + \frac{N}{r}\right)\left(\frac{d_i v}{\nu}\right)^{0.8} \tag{13}$$

where ϕ is some constant to be determined from the experimental data.

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*W. H. McAdams and T. H. Frost, Refrigerating Engineering, Vol. 10, No. 9, March, 1924, p. 323.
†T. E. Stanton, Phil. Trans., Vol. 190A, 1897, p. 67.
‡Webster, Trans. Inst. Eng. and Shipbuilders in Scotland, Vol. 57, p. 58.
§Bray and Sayler, Undergraduate Thesis, Mass. Inst. Tech.

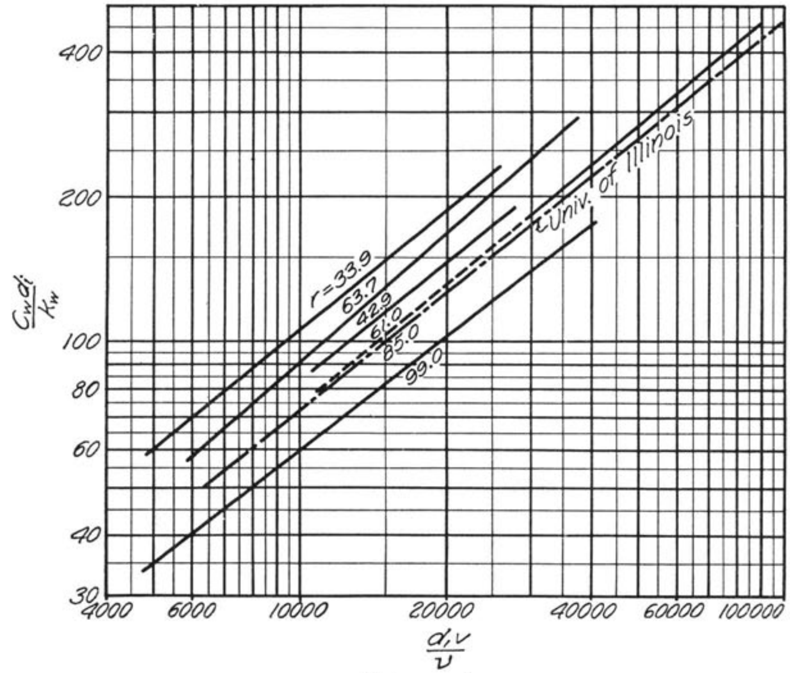

FIG. 20. RELATION BETWEEN $\frac{C_w d_i}{k_w}$ AND $\frac{d_i v}{\nu}$ FOR DIFFERENT RATIOS OF TUBE LENGTH TO TUBE DIAMETER

The entrance conditions have considerable influence on the friction for some distance along the tube. The thickness of film, and hence the conductance, is also a function of the frictional resistance at the surface. It therefore appears reasonable that with comparatively smooth tubes the ratio of tube length to tube diameter r would be a factor largely determining the location of the different curves as shown in Fig. 20. This effect may be analyzed by plotting the values of $\frac{C_w d_i}{k_w}$ for some constant value of $\frac{d_i v}{\nu}$ against the corresponding values of $\frac{1}{r}$. This has been done as indicated in Fig. 21 for $\frac{d_i v}{\nu} =$ 12 380. This line conforms to the one made use of by McAdams and Frost* in their analysis, and has been drawn in the same location relative to the plotted points. The equation of this line is

*Loc. cit.

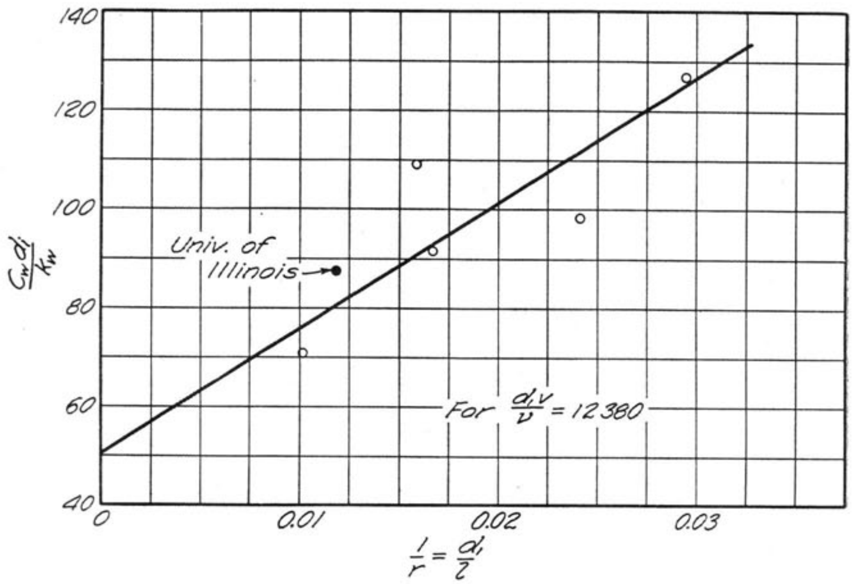

FIG. 21. RELATION BETWEEN $\frac{C_w d_i}{k_w}$ AND $\frac{d_i}{l}$

$$\frac{C_w d_i}{k_w} = 50.9\left(1 + \frac{50}{r}\right) \tag{14}$$

Substituting the value 12 380 in Equation (13) and solving Equations (13) and (14) for $\phi\left(1 + \frac{N}{r}\right)$

$$\phi\left(1 + \frac{N}{r}\right) = 0.02695\left(1 + \frac{50}{r}\right) \tag{15}$$

Equation (13) may now be written

$$C_w = \frac{0.02695\, k_w}{d_i}\left(1 + \frac{50}{r}\right)\left(\frac{d_i v}{\nu}\right)^{0.8} \tag{16}$$

Equation (16) may be used to predict the conductance of the water film based on the area of the water side of the tube. If the value of C'_w is now substituted from Equation (5), Equation (16) becomes

$$C'_w = \frac{0.02695\, k_w}{d_o}\left(1 + \frac{50}{r}\right)\left(\frac{d_i v}{\nu}\right)^{0.8} \tag{17}$$

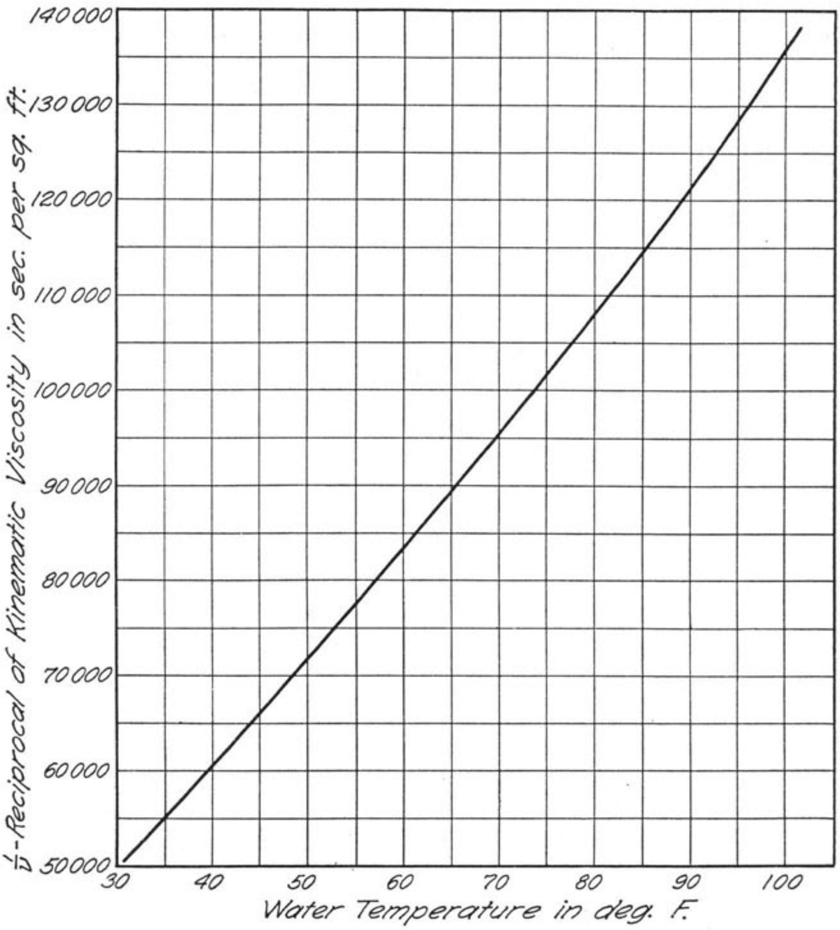

FIG. 22. KINEMATIC VISCOSITY OF WATER FOR DIFFERENT WATER TEMPERATURES

Equation (17) may be used to predict the conductance of the water film based on the condensing surface, or the outside area of the tubes, provided that correct values are assigned to ν and r.

21. *Application of Equations to Test Results For Commercially Clean Tubes.*—In order to apply Equation (17) to a practical problem in the design of a condenser, it would first be necessary to obtain r from the probable dimensions of the condenser, and then to assume an allowable mean temperature for the water. The drop in temperature through the ammonia film and tube could then be calculated

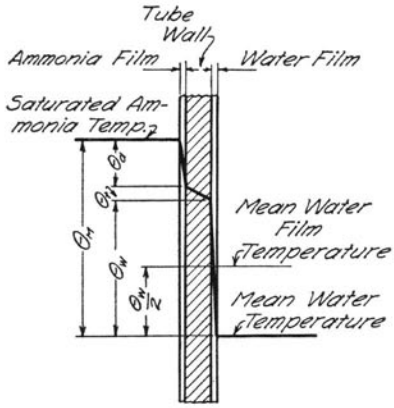

FIG. 23. DIAGRAM OF TEMPERATURE GRADIENT THROUGH TUBE WALLS AND FLUID FILMS

and the temperature of the water film assumed to be midway between that of the water surface of the tubes and the mean temperature of the water. The value of ν could then be obtained from a curve similar to the one shown in Fig. 22,* and the coefficient of heat transfer for a given water velocity, or the water velocity required for a given coefficient of heat transfer, could be calculated.

In the case of the tests on the multitube-multipass condenser, the data for calculating the temperature of the water film were available, and it was desired to determine the applicability of Equation (17) to the actual test results by first calculating the values of C'_w and then adding the constant value, 0.00085, of the intercept in Fig. 19 to the calculated values of $\frac{1}{C'_w}$ to obtain the calculated or predicted values of $\frac{1}{K}$.

Let t_s = temperature of the saturated ammonia, deg. F.
t_w = mean temperature of water film, deg. F.
θ_m = mean temperature difference between saturated ammonia and water in the tube, deg. F.
θ_a = mean temperature drop through ammonia film, deg. F.
θ_t = mean temperature drop through tube wall, deg. F.
θ_w = mean temperature drop through water film, deg. F.

*Hütte, Landolt, and Börnstein Physical-Chemical Tables.

Then, from Fig. 23,

$$\theta_w = \theta_m - (\theta_a + \theta_t) \tag{18}$$

$$t_w = t_s - \theta_m + \frac{\theta_w}{2} \tag{19}$$

$$\therefore \quad t_w = t_s - \left(\frac{\theta_m + \theta_a + \theta_t}{2}\right) \tag{20}$$

The total heat transfer may be calculated from the heat given up by the ammonia

$$H = 60\, W\, (i'' - i') \tag{21}$$

in which H = heat transferred from the ammonia to the water, B.t.u. per hour.

W = ammonia condensed, lb. per min.

i'' = heat content of dry saturated ammonia vapor at the temperature of liquefaction in the condenser, B.t.u. per lb.

i' = heat content of the liquid ammonia at the temperature of the liquid leaving the condenser, B.t.u. per lb.

The temperature drop through the ammonia film may be calculated from

$$\theta_a = \frac{H}{C_a A_o} \tag{22}$$

or substituting from Equation (21), and using the value $C_a = 1635$ as determined from the intercept in Fig. 19

$$\theta_a = \frac{60 W (i'' - i')}{1635 \times 93.8} \tag{23}$$

The temperature drop through the tube may be calculated from

$$\theta_t = \frac{H\, x_t}{k_t A_m} = \frac{0.00795 \times 60\, W\, (i'' - i')}{34.9 \times 89.5} \tag{24}$$

Since the liquid leaving the condenser was at the same temperature as the saturated ammonia, sufficient data are given in Table 1 to permit the calculation of the mean temperature of the water film from Equations (24), (23), and (20). This has been done for each test and the results are also given in Table 1.

The correction factor, $50.9\left(1 + \frac{50}{r}\right)$, which was obtained in Equation (14) to correct for entrance conditions, was derived from investigations on single straight tubes. Just what effect the arrangement of tubes in the multitube-multipass condenser would have on this correction factor is somewhat problematical. In the final analysis, therefore, the application of Equation (17) to the results obtained on these tests must be regarded as a test of the validity of this correction factor as applied to a commercial condenser of the type tested. In the present case the value of r was obtained from the following considerations: Since the two shells were connected in parallel, the friction pressure losses, and hence the entry conditions, for the condenser as a whole were identical with those for a single shell. In a single shell the water passed through the seven tubes in series, and the seven tubes were regarded as a single tube with length $7l$, having the equivalent of seven entry losses. The ratio r for a single tube would be $\frac{12.82}{0.151} = 85$, and the factor $\frac{N}{r}$ was regarded as $\frac{7 \times 50}{7 \times 85} = \frac{50}{85}$, which would conform to the $\frac{50}{r}$ in Equation (17).

From the film temperatures given in Table 1 it is possible to obtain three separate curves for the calculated values of C'_w, one for each of the three condenser pressures, 125, 150, and 175 lb. per sq. in. gage. However, since the points representing the observed values of $\frac{1}{K}$ in Fig. 19 did not appear to be arranged in separate bands corresponding to the three condenser pressures, a single line representing the composite values of $\frac{1}{K}$ was drawn. It is probable that with increased condenser pressures a corresponding decrease in the conductance of the ammonia film occurs to offset the normally increased conductance of the water film resulting from the higher mean temperatures of the water for a given water velocity. Hence, the effect of increased temperature observed in the calculated values of C'_w would not be reflected in the observed values of K or $\frac{1}{K}$. This could not be accounted for in the present calculations, because the values of $\frac{1}{K}$ were obtained by adding the fixed intercept 0.00085, representing the resistance of the commercially clean tubes and ammonia film, to the calculated values of $\frac{1}{C'_w}$. For this reason the water film temperatures given in Table 1 were averaged, and the average value $\frac{1}{\nu}$ = 105 600 read from the curve in Fig. 22, corresponding to the average temperature 78.1 deg. F., was substituted in Equation (17) in order to obtain the single average curve $\frac{1}{C'_w}$ (calculated), which is shown as a broken line in Fig. 19. Also, a constant value of 0.329 B.t.u. per hr. per sq. ft. per deg. F. per ft. of thickness was used for k_t. Substitution in Equation (17) gives

$$C'_w = \frac{0.02695 \times 0.329}{0.1667}\left(1 + \frac{50}{85}\right)(105\,600 \times 0.1508\, v)^{0.8}$$

Therefore,

$$C'_w = 194\, v^{0.8} \tag{25}$$

This line has been shown as $\frac{1}{C'_w}$ (calculated) in Fig. 19, and the equation may be compared with that of the observed line given in Equation (9).

The line $\frac{1}{K}$ (calculated) in Fig. 19 was obtained by adding the intercept 0.00085 to the values calculated from Equation (25), and it may be observed that it also represents the observed values of $\frac{1}{K}$ with a fair degree of accuracy.

It has been indicated that any uncertainty in the application of Equation (17) to commercial condensers lies in the evaluation of the ratio r, and the fact that the correction factor for entrance conditions was obtained from a consideration of data obtained on single straight tubes with one entry loss as shown in Fig. 21. The points representing the data show considerable deviations from the line drawn, and the line so drawn has been made to conform to the one used by McAdams and Frost* in order that the final Equation (17) might conform to the one deduced by them.

If it is assumed that the resistances of the ammonia film and the tube wall remain practically constant over the whole range, and that the line $\frac{1}{C'_w}$ (observed) in Fig. 19 is a correct representation of the

*Loc. cit.

observed values of $\frac{1}{C'_w}$ for the University of Illinois tests, corresponding values of $\frac{C_w d_i}{k_w}$ and $\frac{d_i v}{\nu}$ may be calculated, and a line representing these tests drawn as shown in Fig. 20. From this line another point, as shown in Fig. 21, may be derived. A line drawn through this point would represent the composite data almost as well as the line that was used. Under these conditions a new value for the correction factor $\phi\left(1 + \frac{N}{r}\right)$, given in Equation (13), could be derived, and by using this factor in Equation (17) the line representing the calculated values of $\frac{1}{K}$ in Fig. 19 could have been made identical with the one representing the observed values of $\frac{1}{K}$. It has not been considered advisable, however, to derive a new correction factor and rewrite Equation (17), for the reason that, while it might be representative of tests on the multitube-multipass condenser used, the data obtained from one size of condenser are not sufficient to draw the conclusion that the correction factor so derived would be representative of the multitube-multipass type as a whole. The equation, as it exists in the literature at present, although not based on data including this type, represents the results within limits of about 10 per cent, which is probably an acceptable degree of accuracy for the practical purposes of condenser design. However, the analysis does indicate the advisability of more fundamental work being done using tube arrangements corresponding to those found in commercial types of condensers in order to establish relations similar to those shown in Fig. 21, from which correction factors strictly applicable to the various types may be derived.

22. *Application of Equations to Test Results for Fouled Tubes.*—The results of the GX series of tests indicating the relation between the values of K and the rate at which scale formed on the tubes are shown in Fig. 14.

In the case of fouled tubes Equation (6) must be modified by adding a term, $\frac{1}{C'_s}$, the conductance of the scale based on area of condensing surface. In this case the intercept will represent the sum of the resistances of the ammonia film, the tube wall, and the scale. Hence, if the line for commercially clean tubes has been determined as in Fig. 19, the resistance of the scale may be found by drawing a line

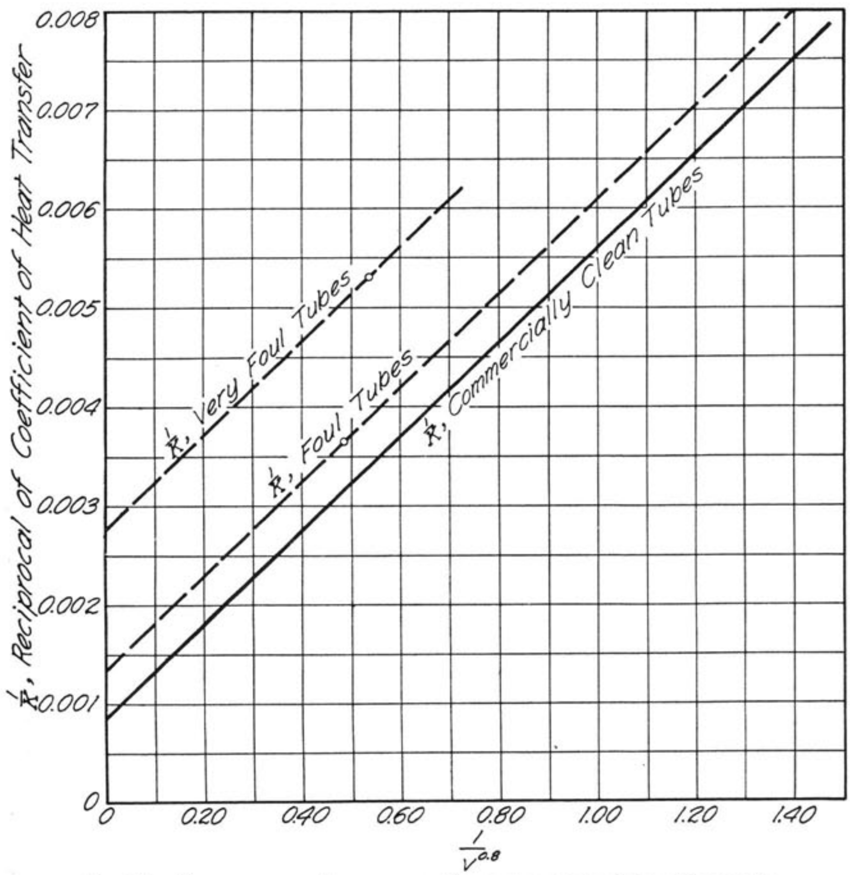

FIG. 24. OBSERVED AND CALCULATED COEFFICIENTS OF HEAT TRANSFER FOR FOULED TUBES

through some value of $\frac{1}{K}$, determined from the fouled tubes, and parallel to the line for $\frac{1}{K}$ for the clean tubes. The difference between the intercepts will be $\frac{1}{C'_s}$.

In Fig. 24 the line for commercially clean tubes shown in Fig. 19 has been reproduced. After 16 hours of continuous running with 331 lb. of water per min., or a water velocity of 2.48 ft. per sec., the value of K from Fig. 14 is 274 B.t.u. per sq. ft. per deg. F. per hr.

The corresponding values of $\frac{1}{K}$ and $\frac{1}{v^{0.8}}$ have been plotted in Fig. 24, and the line $\frac{1}{K}$ for foul tubes has been drawn parallel to the one for clean tubes. The difference in intercepts is $0.00135 - 0.00085 = 0.0005$. Hence the conductance of the scale, C'_s, after 16 hours of running, was 2000 B.t.u. per sq. ft. of condensing surface per deg. F. per hr.

When 209 lb. of water per min. were circulated the velocity was 1.56 ft. per sec. and $\frac{1}{v^{0.8}} = 0.701$. The corresponding value of $\frac{1}{K}$ read from the curve in Fig. 24 is 0.00467, or $K = 214$. This point on the curve for 209 lb. of water per min. in Fig. 14 would give the number of hours necessary to run at this rate in order to build up a scale resistance of $\frac{1}{2000}$ corresponding to that obtained at the 331 lb. rate after 16 hours of running. The time thus obtained is 9.2 hours. These two points have been connected with the line marked "Scale Resistance = $\frac{1}{2000}$" in Fig. 14.

Test XF in Table 1 was run when the tubes had a comparatively thick deposit of scale, while test YF was run under the same conditions of load and water velocity, but with clean tubes. For test XF, $\frac{1}{v^{0.8}} = 0.532$, and $\frac{1}{K} = 0.00535$. This point has also been plotted in Fig. 24, and the line $\frac{1}{K}$ for very fouled tubes has been drawn parallel to the one for clean tubes. The difference in intercepts is $0.00275 - 0.00085 = 0.00190$. Hence C'_s for the very fouled tubes is 527 as compared to 2000 for tubes only moderately fouled.

VI. Conclusions

23. *Conclusions.*—The following conclusions may be drawn from this investigation, subject to such limitations as may be imposed by the range covered, and by the conditions specifically stated under which the tests were run:

(1) For a given initial water temperature and given weight of cooling water per minute the increase in total tonnage is directly proportional to the increase in condenser pressure expressed in lb. per sq. in. gage.

(2) The actual total capacity developed is governed by the amount of condensing surface, the limiting condenser pressure, the temperature of the water available, and the amount of water circulated.

(3) For a given difference between the ammonia saturation temperature and the initial water temperature, the condenser pressure has no material effect on the total tonnage developed with a given weight of water circulated per minute.

(4) For a given difference between the ammonia saturation temperature and the initial water temperature, the increase in tonnage developed is directly proportional to the increase in weight of cooling water circulated per minute.

(5) For given water velocities through the condenser tubes, lower values are obtained for the coefficients of heat transfer when the two identical shells of the condenser are connected in series than when they are connected in parallel.

(6) The most effective arrangement for condensing surface is one in which the water travel is comparatively short, and which offers a large area for the action of the coldest water near the point of entry.

(7) For a given weight of water circulated per minute per square foot a higher value for the coefficient of heat transfer and for the total tonnage developed is obtained when the two identical shells of the condenser are connected in series, than when they are connected in parallel, but these higher values are accompanied by greatly increased friction pressure losses, and hence increased power is required to circulate the cooling water.

(8) The decrease in the heat transmission caused by fouling of the tubes was very marked during the first four hours of running subsequent to cleaning. At the end of this time the effect of scale formation materially decreased and the rate of heat transfer became practically constant over a wide range of water rates.

(9) For a given tonnage developed and a given water rate fouling of the tubes results in greatly increased temperature differences between the saturated ammonia and the cooling water, and in greatly reduced coefficients of heat transfer.

(10) For a given initial water temperature the condenser tonnages developed per square foot of condensing surface over a range of volumes of water circulated per minute per square foot can be represented by a mean curve, with deviations not exceeding 8 per cent for any of the condensers tested. It is possible that these curves may be of more practical value than those for mean coefficients of heat transfer.

(11) The general equations, now existing in the literature, for heat flow in condensers may be used within limits of 10 per cent to predict the coefficients of heat transfer obtained for the multitube-multipass ammonia condenser under the conditions and over the range of water velocities used in these tests.

(12) More fundamental work should be done with tube arrangements corresponding to those found in commercial types of condensers in order to establish factors in the equations to render them more generally and more strictly applicable to the various types.

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